COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 9, No. 10 APRIL, 1938 25c a copy

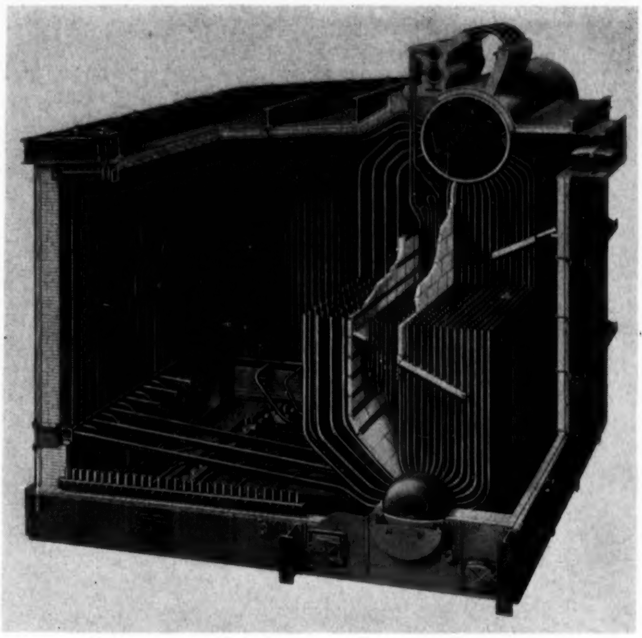
Recently completed Type VU steam generator of 40,000 lb per hr capacity with bowl mill

High-Pressure Centrifugal Boiler Feed Pumps

Present Trends of Industrial Power Plant Practice

Estimation of Radiant Heat Exchange in Boiler Furnaces

6 reasons for the FINE PERFORMANCE of the Type VU Steam Generator . . .



In the selection of steam generating equipment most considerations, even price, are, in the last analysis, secondary to efficient, dependable performance and trouble-free operation. In order to assure these results the means to obtain them must be inherent in the design and construction of the equipment.

The C-E Steam Generator, Type VU, has numerous superior design and construction features which largely account for the exceptional results being obtained with this unit in all parts of the country. Among the more important of these features, which merit special consideration on the part of prospective purchasers of steam generating equipment, are the following:

1 All of the most active steam producing tubes enter the drum above the normal water line and discharge equally across the full length of the drum. This arrangement reduces turbulence to a minimum and permits effective utilization of the entire steam release space. Dry steam and a stabilized water level are thus assured even under adverse conditions.

2 All pressure parts of the boiler and furnace are suspended from an independent steel structure which also acts as a frame for the steel casing. Destructive stresses and sealing problems resulting from expansion are entirely eliminated by this design.

3 The location and type of burners assure effective utilization of the furnace volume. Products of combustion enter the tube bank uniformly across its full width at low velocity, a condition requisite to the prevention of troublesome slag formation on the tubes in this area.

4 The unit is of symmetric design in that any section taken through the unit from front to rear is similar to any other section so taken. The gases enter the tube bank at a practically constant temperature across the width of the unit and this characteristic is maintained as they progress over the remaining heating surfaces. The conventional baffle arrangement used directs the gases effectively over the entire heating surface assuring maximum heat absorption with minimum draft loss.

5 The furnace walls are fully protected with water walls backed up successively with special tile, refractory and insulation with an outer covering of steel panels. This is a distinctly superior type of wall construction. It minimizes heat loss from radiation and air infiltration and virtually eliminates setting maintenance.

6 A double row of water screen tubes with the side wall circulators below them provide a relatively cool zone in which the descending ash particles are converted into a dry, easily-removable state. There is no ash accumulation in the combustion zone.

A brief study of the sectional view above will provide a better appreciation of these features and the many others which distinguish the VU Generator.

The VU Steam Generator is available in units ranging from 30,000 to 250,000 lb of steam per hr capacity. We invite your comparative analysis of its features, point for point, with any equipment of the same general type now on the market. We shall be pleased to send you a catalog and supply any additional data desired.

C·E

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ALL TYPES OF BOILERS, FURNACES, PULVERIZED FUEL SYSTEMS, STOKERS; ALSO SUPERHEATERS, AIR HEATERS AND ECONOMIZERS

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME NINE NUMBER TEN

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H. STUART ACHESON, *Advertising Manager*

ALFRED D. BLAKE, *Editor*

THOMAS E. HANLEY, *Circulation Manager*

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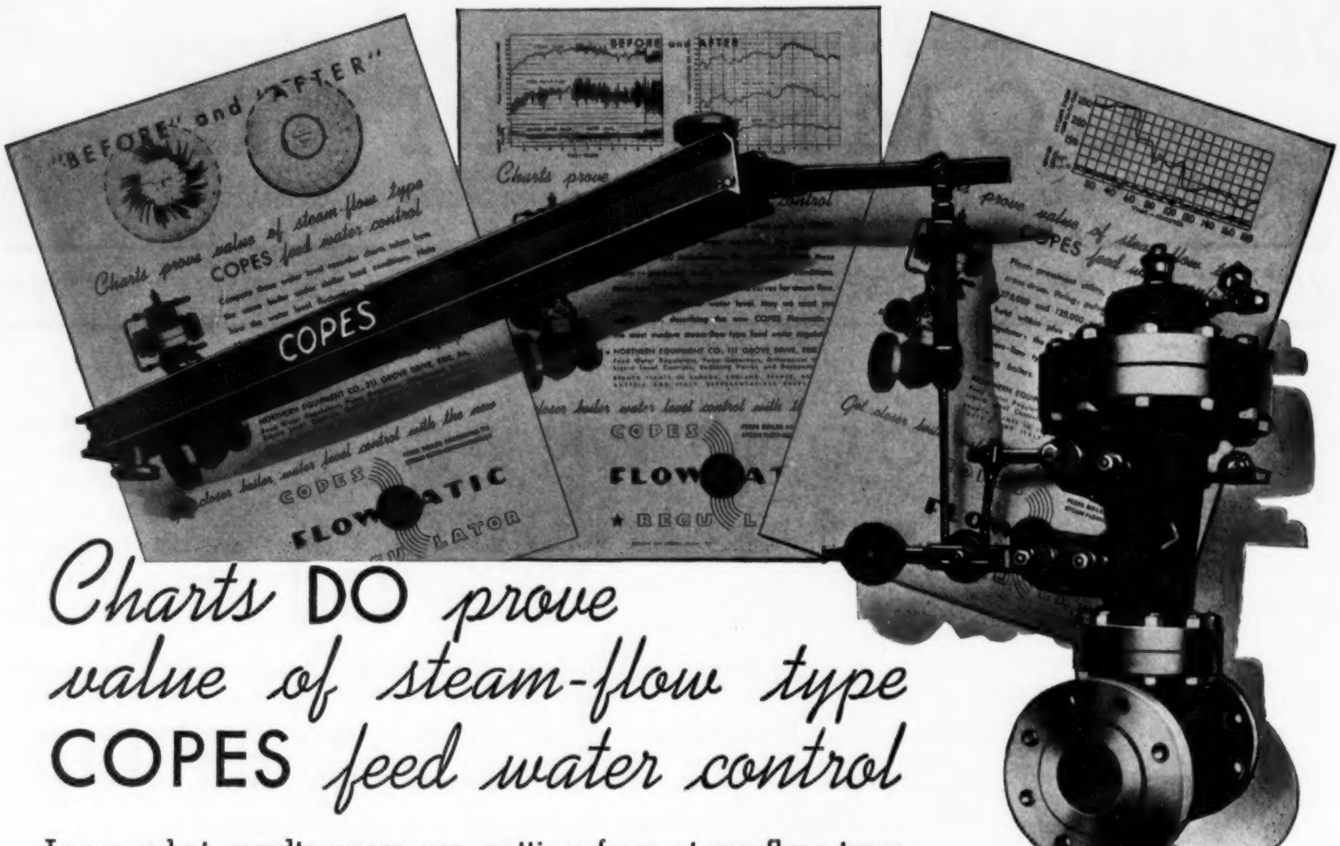

COPES

EDITORIAL

Licensing of Patents

Pending before Congress is the McFarlane Bill, HR-9259, pertaining to the licensing of patents. This, in effect, would shorten the exclusive life of a patent to the patentee from seventeen to three years by making it incumbent on the Commissioner of Patents, after three years from date of issuance, to grant a license for its use, under royalty, to any responsible applicant. The royalty, not to be less than one-half of one per cent nor more than ten per cent of the cost of manufacturing the article or device, would be fixed by the Commissioner of Patents. Appeals from his decision, as to amount of royalty *only*, could be made to a Board of Appeals, to be appointed by the President, or from that body to a District Court.

The purpose of the bill is to supplement the anti-trust laws by making it impossible for an individual or manufacturer to enjoy a monopoly of any design or device in a competitive field and to prevent the shelving or non-use of patents. Actually it would be inimical to our patent system and discourage inventive genius.

In establishing the American patent system many years ago our forefathers recognized that progress could be stimulated by making possible a reward to those who could produce new and useful inventions. Seventeen years was fixed as a reasonable period in which to perfect an invention, to finance it and to build up a consumer demand, with an opportunity thus to recompense the inventor for expenses incurred in development and to permit a profit commensurate with the effort involved. Practically all foreign countries have followed this course, the periods ranging from fifteen to twenty years.

Without this protection there would be little incentive for invention and development because of the time and expense usually required to perfect a new device. The royalties prescribed in the pending bill take no cognizance of development or marketing costs and apply only to manufacturing costs which in many cases might be insignificant compared with the value of the product. Thus the inventor would be left "holding the bag" while his competitors reaped a profit from his efforts. In this connection, it is conceivable that certain competitors, because of their manufacturing facilities and marketing setup, would be able to dominate the field and thus defeat the primary purpose of the bill.

Enactment of such legislation would work an injustice to those who have spent years and fortunes in perfecting inventions that have resulted in public benefits, as is attested by the history of numerous well-known inventions. It would also stifle development and industrial progress. While it might prove a boon to the legal profession by incurring endless litigation, it would undoubtedly cause much unemployment among research workers, and, by retarding invention, would affect employment in those fields that are stimulated by new developments. Furthermore, it would centralize power in government officials who might be poorly qualified to evaluate the situation in many cases.

The bill is being actively opposed by the New York Patent Law Association, the Cleveland Patent Law Association and many similar bodies, as well as inventors and small business enterprises. Latest advices from Washington indicate that the opposition is becoming so formidable as to lessen the chances of the bill's passage at this session of Congress, despite the fact that it is understood to be on the list of *must* legislation prescribed by the Administration.

Teachers Accused of Bias

In a recent talk before the League for Industrial Democracy, Morris L. Cooke, former head of the Rural Electrification Administration, is reported as having charged that engineering students are becoming biased against public ownership due to the influence of their professors who are subsidized through vacation jobs with private utilities and large manufacturers of power-generating and electrical equipment.

This is a serious accusation to make unless supported by full and convincing evidence and is an inditement of the independence of the teaching profession.

It is true that many instructors do take summer jobs of this character in order that they may keep abreast of practice and rationalize the book teachings with what is actually going on in the field. In so doing they not only become better teachers but the students become better fitted to take their place in industry upon graduation. The companies, on the other hand, are usually willing to cooperate with educational institutions solely with the foregoing objective. If there be some with such ulterior motives as suggested by Mr. Cooke, they are indeed in the minority.

It is generally conceded that, to warrant serious consideration, accusations of bias should emanate from a totally unbiased source.

The TVA Investigation

As this issue goes to press there appears to be some difficulty in completing the list of senators to serve on the Joint Congressional Committee that will investigate the TVA, the excuse being the probable time involved in the hearings because of their contemplated broad scope. It should not take very long to ascertain whether the charges made by former Chairman Arthur Morgan are well founded and whether there has been proper accounting for the vast expenditures, but the decision to extend the investigation to numerous other phases of the project will likely cause the hearings to drag out for many months. Among these, the inclusion of activities on the part of utilities in opposing the TVA can have little effect on the points at issue, but will probably serve as a useful smoke screen.

High-Pressure Centrifugal BOILER FEED PUMPS

By A. H. RICHARDS, Field Engineer
INGERSOLL-RAND COMPANY, Cameron Engineering Service

With design, manufacture and operation forming the three essentials to satisfactory performance, the author discusses briefly the importance of metallurgy in design and precision in manufacture and stresses the necessity for intelligent operation. In the latter connection, clearances, packing, protection against the effects of abnormal operating conditions and the necessity of determining and maintaining proper feedwater conditioning are covered. Complete cooperation between user and manufacturer before and after installation is shown to be most important.

A GOOD boiler-feed pump installation depends upon three fundamentals, namely, design, manufacture and operation. If any one of these is slighted, disastrous consequences soon follow. This applies equally to low-pressure or to high-pressure installations, the only difference being that the higher the pressure the more quickly will shortcomings show up.

It is not the purpose of this article to describe in detail the features of design and manufacture, but rather to discuss operation in some detail. Regardless of how well designed or carefully manufactured a pump may be, faulty operation can ruin an installation within a very few minutes. It is proposed to show that a pump installation is entirely dependent upon operation, once the other two factors, design and manufacture, have been properly adjusted; and that the longer an installation is in actual service the more secure it becomes on this *three-legged* foundation. Operation is the last leg added and oftentimes is extremely wobbly and, unless the proper bracing in the form of education is added, this last leg may cause the collapse of the whole works.

Design Features

With the mushroom growth of topping units, boiler-feed pump manufacturers were faced with serious problems of design, more severe than anything previously encountered; with inquiries originating from competent engineers and operators who had spent much time investigating older high-pressure installations and who were as familiar as the manufacturer with troubles that had occurred in the older installations. Those manufacturers who could rely on past experience in high-pressure work were able to avoid the more obvious pitfalls, but the engineers are always thinking up new problems for the manufacturer to improve the operation of boiler-feed pumps and to insure the maximum reliability. Problems of stuffing box arrangements, materials, pump protection at light loads, temperature variations, suction heads, etc., all change with each installation. Arrangements that work in one installation will not necessarily be fitted to another.

Below are listed the pump service conditions of a few of the high-pressure plants in order to give some idea of the scope of the work required of modern high-pressure boiler-feed pumps:

Capacity, Gpm	Suction Pressure, Lb per Sq In.	Discharge Pressure, Lb per Sq In.	Water Temp, Deg F	RPM
1550	400	1800	338	3575
850	260	1630	390	3550
535	115	1550	338	3550
1025	345	1645	400	3450
870	69	1655	300	3575
850	20	1635	220	3550
1150	325	1670	380	3550
1000	20	1795	228	3575
255	250	1800	250	6000
1600	600	1600	440	3575
1330	700	1600	395	3575
800	50	1600	70	3600
1830	40	1640	150	3470
1440	550	1650	430	3450
1500	250	1600	325	3400
800	350	1500	210	3550

The purchaser of high-pressure boiler feed pumps should insist upon the pump manufacturer having an intimate and up-to-date knowledge of materials used in the manufacture of pumps. Successful modern design depends upon the proper use of materials for low maintenance operation. There is a specific material for every part of a high-pressure pump and the selection of this material should be made by competent metallurgists and not from advertising circulars. The metallurgist must work in close relationship to both the design and manufacturing divisions and be intimately familiar with the actual work performed by the various parts of the pump. With the same basic material, widely varying performance in a pump is obtained by a proper knowledge and control of heat treatments. It is as important to know when not to heat treat as it is when to heat treat. Strain relieving anneal for relief of residual stresses in materials plays a most important part in selection and use of materials. What parts may be suitably manufactured from castings or forgings or by welding are vital functions of the metallurgist's job in modern pump manufacture. Often the use of a material will be indicated from every standpoint of design or manufacture, in which case field experience with this material under similar conditions should be carefully checked. The proposed feedwater treatment and its

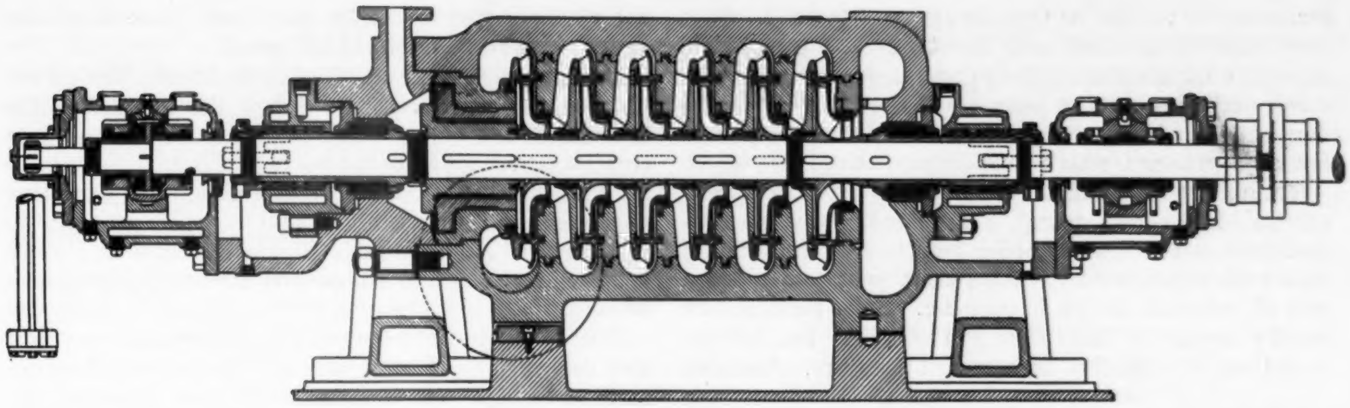


Fig. 1—Cross-section of typical high-pressure boiler feed pump

control should be made known to the pump manufacturer to assist in the proper selection of materials.

The basic design features of successful high-pressure boiler-feed pumps are very similar. Externally, they may be of the horizontally-split type or of the barrel type. Internally, they are almost universally of the single-suction impeller arrangement, employing a separate balancing device to compensate for hydraulic thrust. A typical cross-section of such a pump is shown in Fig. 1 and a photograph of the finished pump is shown in Fig. 2. A photograph of a barrel-type pump of essentially the same basic internal design is shown in Fig. 3.

Stuffing box arrangement forms a major problem in design, particularly from an operating viewpoint. By referring to the list of conditions previously given, it will be noted that suction pressures vary from 20 to 700 lb per sq in. and that temperatures increase with the pressure. The rubbing speed of the shaft sleeve within the packing approaches 5000 ft per min or approximately 80 ft per sec. Most packing materials that will withstand the range of pressures and temperatures are basically frictional materials, their chief ingredient being asbestos, possibly covered with aluminum foil. Depending on the exact conditions, a lead or babbitt-covered asbestos may be used but such high rubbing speeds readily generate a melting heat if the packing is improperly adjusted. From a pump operating standpoint, a reduction of pressure ahead of the packing is highly desirable. This is usually done by permitting a small amount of water to leak out between a rotating and stationary part to an outside source at lower pressure. Such procedure at times causes some slight upset in heat balance within the station but the use of such a bleed-off may be the lesser of two evils. It is definitely easier for the operators to adjust and maintain packing under low pressure than it is under high pressure, and considerable experience and skill are required of the operator to prevent damage to shaft sleeves or frequent pump shut-downs when stuffing boxes are subject to high pressures. It is highly desirable that the stuffing box arrangements be made a point of discussion between the purchaser and supplier before the station details have been finally settled. Each installation is a separate problem requiring careful analysis.

The design clearances used in boiler-feed pumps, between rotating and stationary parts, should be based on experience, shop tests and materials used. An attempt will be made to show later that with proper operation the above points govern the safe minimum clearances; and that with improper operation the minimum safe clearances must be so large as to be impractical.

There is one advantage to the manufacturer of a 1500-lb pressure pump and that is no purchaser will consider cast-iron casings for such pressures. The present ASA flange standards are all based on steel fittings with raised faces and when a manufacturer sells a pump with a cast-iron casing he is compelled to use a steel flange standard with a raised face for a cast-iron flange. Most piping contractors make up the joints using the same technique

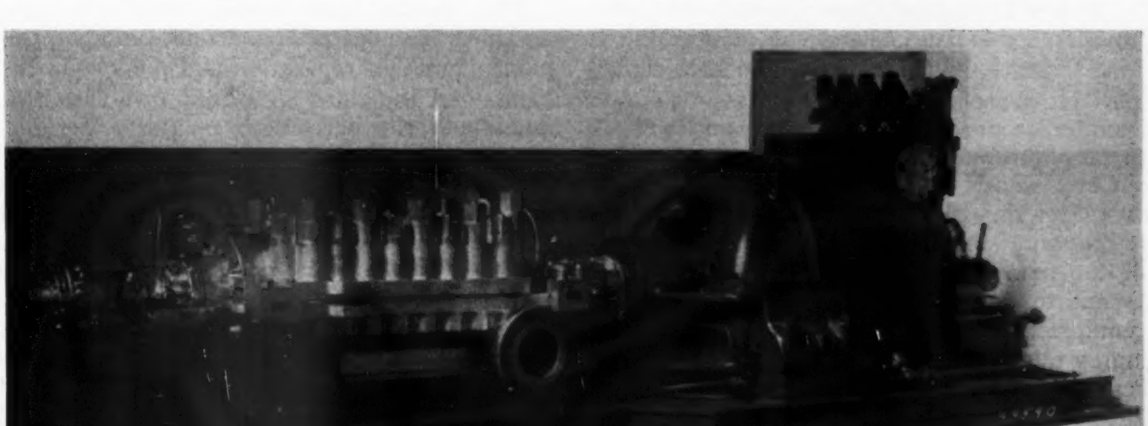


Fig. 2—Boiler feed pump of split-casing type designed to feed a 1400-lb boiler

for cast-iron to steel as they do for steel to steel. This puts stresses of eight and ten times the permissible amount on the cast-iron flange and the result is often a crack occurring at the junction of the nozzle and the flange. The obvious remedy is to use steel casing pumps for any pressure where steel pipe fittings are used.

Variable speed forms an ideal type of drive for a centrifugal boiler-feed pump. It permits variable capacity from the lightest boiler load to maximum boiler load with approximately 20 per cent variation in speed and at constant discharge pressure. Such performance exactly meets the demand from the boiler; i.e., infinite variations in capacity at constant pressure. Variable speed drive also lessens maintenance and prolongs the life of governing equipment and valves in the feedwater system, since constant excess pressure is maintained without throttling action through the valves.

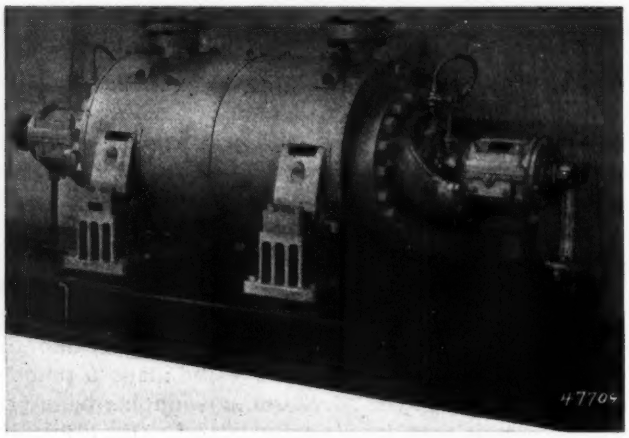


Fig. 3—Exterior view of barrel-type pump

Mounting of boiler-feed pumps is of great importance because of the varying temperatures to which any given installation will be subjected. Despite opinions to the contrary, flexible couplings transmitting 1500 to 2000 hp at 3600 rpm, or above, will not compensate for extreme misalignment with vibration-free operation. The floating or jackshaft coupling arrangement is not used to permit misalignment but to allow easy access to the driven end of the pump to simplify inspection and shaft sleeve maintenance. Misalignment results in vibration; hence the manufacturer should make adequate provision for expansion and contraction due to temperature changes. Such provisions include centerline support on cooled cradles, doweling at one end so that expansion occurs in one direction, and suitable guides to keep the pump shaft in constant alignment with the driver shaft. The same provisions should be insisted upon for the driver as alignment is a dual job.

No attempt will be made here to discuss the detailed design of the pump. The manufacturer's experience, reliability and test equipment should govern the selection of equipment rather than designs on paper not backed up by adequate test facilities and experience to prove the design.

Precision in Manufacture

A successful high-pressure boiler-feed pump must be manufactured with all the skill of a precision watch. Close tolerances must be maintained and backed up by rigid inspection of machined parts before and during assembly. After assembly, the entire unit should be proven by a running test at full speed.

During the various stages of manufacture, the metallurgist again must carefully follow the materials. He determines the proper stage at which the various heat treatments are given to the parts. It is also his duty to see that materials desirable from a design and operating viewpoint can be machined and respond properly to treatment when indicated. Rigid inspection at various stages of manufacture also assures the finished product's success.

There has been tremendous progress made in machine tool design in the last decade and the successful manufacturer of high-pressure pumps will have taken full advantage of this progress by establishing a replacement schedule of machine tool equipment which will replace old machines with more accurate and more flexible new ones. Many of the modern materials highly desirable in high-pressure pumps cannot be properly machined except by new and modern machine tools, particularly since many of these materials must be accurately ground within extremely close limits. The best materials are dense and tough and a tooled finish will be rough and inaccurate; hence the use of such machining methods will reflect throughout the entire building of the pump and necessitate large tolerances and sloppy fits. Precision work requires precision tools.

The test pit is the proving ground of a high-pressure pump. A properly designed and operated Test Department is an essential part of high-pressure pump manufacture. A running test at full speed with cold water subjects the completed pump to pressures much higher than will be encountered in normal operation. Performance of the pump at full speed should be accurately determined as there may be a vast difference in performance at full speed and at half speed. Hydrostatic tests are of value only to determine soundness of joints or castings and will be of no benefit in determining stability, balance, smoothness and output of a pump. The purchaser of a high-pressure pump has every right to expect that the manufacturer selected will be able to demonstrate the pump performance at full speed and pressure and should so stipulate in the specifications.

Operation

The careful manufacturer who has satisfactorily designed, manufactured and tested a high-pressure feed pump places his reputation and product in the buyer's hands and the third leg of our installation is started. Sometimes the purchaser's operating personnel is inexperienced in high-pressure operation and it is a very human and natural reaction to be more concerned about a boiler or more expensive turbine-generator than it is the lowly feed pump. Someone has called a boiler-feed pump "the heart of the power plant." It pumps the life-blood of the machines through the system to enable the necessary production of steam and eventually electricity. Just as we humans often thoughtlessly overtax our own hearts, thus frequently are the "hearts" of power plants abused.

In the old low-pressure slow-speed days of feed pumps, the physical quantities of mass, energy and time were of small proportions. Today, the relations between these same quantities is such that a successful high-pressure feed pump must be as intelligently operated as a turbine;

yet its demands on the operators are actually very few and extremely simple. In their simplest form, they resolve into three easily remembered rules when the pump is operating:

1. Always keep water flowing into the pump.
2. Always keep water flowing out of the pump.
3. Never suddenly subject a pump to extreme temperature changes.

If these three short rules are followed religiously, a pump that has been properly designed and built will lead to a successful installation. Rules 1 and 2 are interdependent. Naturally, if the water flashes into steam when coming into the pump, it will be impossible to keep water flowing out of the pump. However, it is possible and has happened that water has been flowing into the pump but steam was generated within the pump.

In any centrifugal pump irrespective of design, the difference between the work or horsepower put into the pump and the useful work performed by the pump is converted to heat and must be dissipated. For the purpose of this article, it is proposed to neglect the very small percentage absorbed by the bearings, coupling, windage and stuffing boxes and to make the practical assumption that the water passing through the pump absorbs all of this heat. This can be readily demonstrated by actual tests. Thus, if we calculate the brake horsepower (input) against the generated or water horsepower (output) at various capacities and convert their difference to Btu, we will obtain the quantity of heat added to the water in passing through the pump. By dividing these quantities by the water flowing in pounds, the temperature rise of the liquid is found.

Fig. 4 shows a typical brake horsepower–water horsepower curve for a high-pressure pump and Fig. 5 shows the temperature-rise curve plotted from the differences

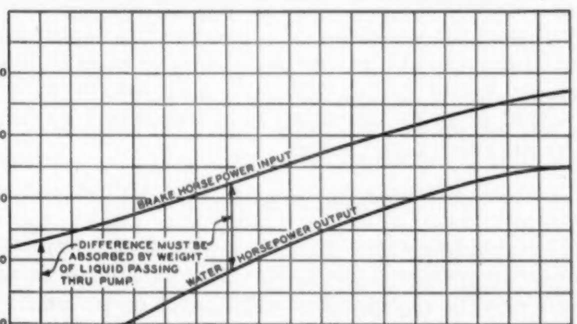


Fig. 4—Typical input-output curves for high-pressure pump

shown in Fig. 4. It will be obvious that with any reasonable flow through a pump the weight of water is sufficient to absorb the difference between brake horsepower and water horsepower without appreciable temperature rise. We, thus, are concerned purely with operation under abnormal conditions and at extremely light loads. An interpretation of abnormal operation is any operating condition which subjects equipment to conditions for which it was not designed and which may have injurious effects on the equipment. These conditions may be brought about by ignorance, by mechanical failure of related equipment, by improper operation of related or interdependent equipment and by natural causes such as thunderstorms, etc. From a pump operating viewpoint, provision must be made either to prevent these abnormal operating conditions, which would indeed be Utopia, or so to protect the pump that despite any abnormal condition no damage to the pump will result. This can be done quite simply.

First, just what harm will result from an abnormal operating condition that results in either zero discharge

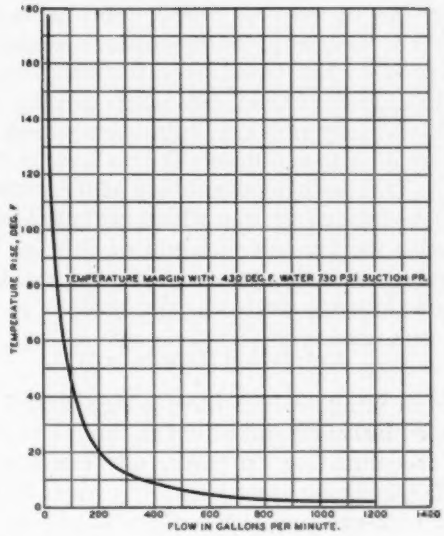


Fig. 5—Temperature rise curve with 430 F water and 730 lb per sq in. suction pressure

from the pump or reduces the flow to a very small quantity? The temperature of the water passing through the pump will be increased as shown in Fig. 5 and this increase will take place in approximately one minute or less. The first result will be an extreme temperature change which may cause the internal rotating parts to heat so much more rapidly than the outer and heavier stationary parts that severe rubbing will occur. It is also possible that renewable stationary parts fitted into larger and heavier stationary parts will be heated so much more rapidly than the larger parts that they will, in attempting to expand, be stressed beyond their elastic limit and actually shrink against the rotating parts.

If the pump is designed with a separate form of balancing device, the liquid passing over this device has passed through each stage of the pump and absorbed heat from each stage. In passing the balancing device, its pressure is lowered to approximately suction pressure and since the liquid has been heated in passing through the pump it tends to flash in the balancing chamber or balancing line.

In Fig. 5, there has been plotted the "Temperature Margin" for an assumed installation. This margin is arrived at by assuming suction pressure in the balancing chamber or outlet of the balancing device. The temperature corresponding to this suction pressure is found from the steam tables and the difference between this temperature and the inlet temperature of water coming to the pump is known as "Temperature Margin." For instance, in the example given in Fig. 5, we assume 430 F water coming to the pump at 730 lb per sq in. gage. The temperature corresponding to this pressure is approxi-

mately 510 F. The difference between 510 F and 430 F is 80 deg; or the installation has an 80 deg Temperature Margin. This means the temperature of liquid passing through the pump may rise 80 deg before flashing occurs in the balancing outlet.

The result of flashing in the balancing device is extremely rapid wear due to vibration and internal rubbing caused by the vibration. Basically, this is due to a combination of incidents happening in an exceedingly small period of time, usually within one or two minutes. The mixture of steam and water will not provide proper cooling for the surfaces or proper lubrication for these surfaces. The flashing probably occurs at various points in the passage of the liquid over the balancing device resulting in a series of small explosions causing the shaft to oscillate violently which, in turn, causes more rubbing of the internal parts and upsets the packing in the stuffing boxes. The mechanical rubbing of the parts creates a tremendous heat of friction which heats the liquid still more because the amount of liquid or flow of liquid is insufficient to cool the parts. This causes still more flashing and the cycle starts all over again.

Rebuilding Expensive

Parts from pumps which have gone through this experience are blackened and pitted as though burned by a torch. For estimating purposes, one can figure on a cost of $2000 as being approximately correct for rebuilding a large high-pressure pump which has been operated at too light a load.

On multi-stage pumps which do not employ a separate balancing device, the situation is a little different and the results may be even more disastrous. Pumps without a balancing device have no place after the first stage where the pressure is reduced to suction pressure. Consequently, once the liquid, as a liquid, is past the first stage, no flashing will occur beyond the first stage, provided no flashing occurs in the first stage. The sudden heating, however, will cause the same distortion as outlined previously, with perhaps worse results due to the more complicated shapes of the casings of such pumps. At extremely light loads, however, flashing will occur in the eye of the first-stage impeller due to recirculation through the rings of the first-stage impeller. When this occurs, the pressure immediately drops at the first stage and this loss of pressure is transmitted through the entire pump, resulting in flashing throughout the pump. The pressure drops in the first stage when flashing occurs because a centrifugal pump will not generate pressure when handling a vapor or a mixture of water and vapor, as it will when handling a solid column of liquid. This flashing in the suction side of the first-stage impeller breaks the column of liquid and results in a pressure drop all the way through the first stage, which is transmitted to the inlet of the second stage where the flashing is even more thorough because the water is progressively hotter from stage to stage through the pump. This, in effect, is the same thing as loss of suction and the lack of liquid to lubricate and cool the surfaces usually results in seizure of the rotating parts. The cost of the damage will probably be about the same or slightly greater than that for pumps using balancing devices due to the possibility of complete seizure taking place.

The foregoing description roughly covers the results of improper or abnormal operation without protective devices. The cure, of course, is prevention. It has been said that pump manufacturers build pumps with too close clearances in an attempt to gain the last point in efficiency; the thought being that an increase in clearances would enable a pump to be operated under abnormal flow conditions without damage. Let us examine the facts carefully and see if this is correct.

Referring to Fig. 4, the curve of water horsepower output is the useful work done by the pump. The ratio of this water horsepower output to the brake horsepower input is the efficiency of the pump. An increase in clearances through the pump will not change the water horsepower in any way, but will increase the brake horsepower because a greater quantity or weight of liquid must be pumped for the same amount of useful output. The first result, then, of increased clearances and lowered efficiency is to increase the amount of heat to be absorbed by the same amount of liquid passing through and out of the pump. Graphically, this will increase the temperature rise shown in Fig. 5 for the same flow; and flashing will occur at higher loads than if the pump were built with normal clearances. The second result of greater clearances will be to cause more extreme temperature changes with the effects already described. It presumably is possible at least to discuss building a pump with clearances such that it could be run so as to generate steam without damage. This has often been done on low-pressure pumps and is entirely practical in certain applications where the pressure and horsepower involved are low. A 50 per cent increase in brake horsepower required on a 25-hp pump would still demand only a 40-hp motor and the additional power would never be found in the auxiliary power cost. A 50 per cent increase in brake horsepower required on a normal 2000-hp pump installation is an appreciable item of cost in auxiliary power, not only in operation but in first cost also. Summing up, a slight increase in clearance simply aggravates the likelihood of damage at light loads and to build high-pressure pumps with such clearances that no damage would occur with improper operation is impractical from an operation cost standpoint as long as there are other alternatives.

Sufficient Flow Important

The simplest remedy is to insure that, despite any abnormal operating condition other than loss of water coming to the pump, there will always be sufficient flow through the pump to insure its safe operation. A small line taken from the discharge and equipped with an orifice nipple will insure a sufficient amount of flow. The orifice nipple serves the dual purpose of reducing pressure and controlling the amount of flow. The required amount of flow will vary with each installation and the most accurate way to handle this point is to plot the temperature rise curve and temperature margin for each installation as shown in Fig. 5. The bypass nipple should then be designed for a somewhat larger flow than is shown by the intersection in Fig. 5. This bypass line should not be taken back to the suction line but should discharge back in the system at some point where the water will flash down to a lower temperature before being returned to the pump, such as the steam side of a heater.

While the bypass arrangement will protect the pump,

it is, of course, useless unless it be open at the time of abnormal operating conditions. The simplest but not the most foolproof method of bypass control is manual control by the operators. However, one failure of an operator to be at the exact spot at the exact time to open the bypass may result in damage and cause a major plant shutdown. If manual control is elected, then the bypass valve should be locked open for the first few months of operation until all the preliminary operating "bugs" have been worked out and regular commercial operation is in force. After such a time has been reached, the bypass line should be opened:

(1) When putting on or taking off a pump.

(2) When changing pumps (the bypass from each pump should be open).

(3) When putting the plant on the line or when taking it off the line.

(4) When the pump load is likely to drop below the safe minimum established by the temperature rise curve of the pump.

If the bypasses are arranged in a header, care should be taken to make sure the header is sufficiently large so that one pump discharging into the bypass at a high pressure will not build up pressure in the bypass line and back off the bypass line a pump that is operating at lower pressure. This occasionally happens when a motor-driven unit operates with a turbine-driven unit.

If a surer method of bypass control is desired, there are a number of different arrangements which should be commercially available but, unfortunately, are not. This appears to be due largely to a lack of understanding of the problem by manufacturers of control equipment and somewhat to the apparently high cost of control equipment. The term "apparently" is here justified because a single operation of the control at a time when manual operation would have failed will economically justify the entire cost of the control equipment. Since the major cause of disaster is one of flow, the operation of an automatic bypass control valve should certainly be controlled by flow. If a flowmeter is installed in the suction side of the pump, an auxiliary tap can be taken from this flowmeter to actuate a valve in the bypass line. At flows below a safe minimum, the bypass valve should open and stay open until flow has reached a value above the minimum. This leads to a simple off-and-on control which should function satisfactorily. It is at present being tried in a few stations but insufficient operating experience prohibits any report on it. The bypass line may also be actuated by a solenoid valve which, in turn, is tripped by impulses from the differential leg across the flowmeter. In either type, the valve in the bypass line should be set to open against failure of the actuating medium; if the air or oil or power fail, the bypass valve should open.

A second possible method of control is by temperature differential. The chief objection to this is the variation in temperature at the pump suction. The writer knows of no installations attempting to govern by use of differential temperature control but believes that a suitable control can probably be worked out. However, it must be kept in mind that the control should operate only on temperature rise through a pump, irrespective of the incoming temperature.

Where pumps equipped with a separate balancing device take suction from a deaerating heater or a direct-contact heater, a most satisfactory bypass is afforded by means of returning the balancing leakage at low pressure direct to the heater where it will flash down to the temperature in the heater. In many installations, the amount of this bypass will be sufficient to insure safe operation with the discharge closed and it will be impossible to operate the pump with the balancing line closed. It may be necessary, depending on the temperature margin available, to supplement the leakage from the balancing device with an additional bypass from the discharge. In these cases, it is still advantageous to use the balancing device leakage by returning it to the deaerating or direct-contact heater, as it is an absolute guarantee that the pump can never be run against shut-off or with zero discharge. Where closed heaters are used, the bypass must be taken from the discharge.

Bypass Regarded as Insurance

Occasionally, sales are promoted by involved discussions of pumps requiring or not requiring bypasses and as to whether one type of pump requires more water to be bypassed than another. However, the natural laws involved affect all pumps and a bypass is just so much insurance. Its use does not require additional pump capacity because it is never used at times when the pump is operating at or near its full capacity. It does not add to the cost of operation because at the low capacities where it is used the brake horsepower curve is so flat that the slight additional amount of water used through the bypass will hardly affect the power demand. In addition, if the generating unit is down to such low loads that the bypass on a boiler-feed pump must be open, it is no time to be concerned with feed pump efficiency.

So much has been said and written in the past on the dangers of loss of water going to boiler-feed pumps that troubles rarely if ever occur from this source. Almost all operators thoroughly understand this danger and designing engineers usually plan two or more emergency sources of suction. It is common practice to install level or pressure controllers in most deaerating heaters and usually a low-water alarm is also installed. When the main boiler-feed pumps take suction from a set of primary pumps, it requires careful design planning to insure that, no matter what happens, the primary pumps will continue to feed the main pump suctions or else the main pumps should be arranged to shut down automatically. Obviously, if there is no water going to the main feed pumps, they will be unable to feed the boiler and unless they are automatically and immediately shut down the pumps will seize and destroy themselves. The fires must be pulled under the boiler in any event and the pumping equipment might as well be saved so the unit can be fired up again when normal flow to the pumps is established. Seizure will probably occur in a matter of seconds, assuming complete loss of water. With only partial loss of water, the pump discharge pressure will drop and the check valve presumably close, so the pump will operate either at shut-off or on its bypass. The seizure will be caused by heat of friction of the rotating parts in the vapor or stagnant water mixture causing sudden expansion of the rotating parts. These contact the stationary parts and the heat of friction is instantly increased to the point where complete galling and occasionally actual fusing of the parts

take place. The remedy is again prevention to make sure that water in liquid form is coming to the pump at all times or else to automatically and immediately shut down the pump.

Where boiler feed pumps take suction directly from deaerating heaters it is well to insure that there will be no sudden lowering of pressures within the deaerating heater. If sudden lowering of heater pressure occurs, there may be almost normal liquid level maintained within the heater but the sudden pressure drop creates a cone of vapor bubbles in the suction line which may possibly be carried into the pump, causing a momentary loss of suction. If the pressure drop is not extreme and the period of partial flashing lasts only for a few seconds, no damage is likely to occur to the pump. With high-speed, high horsepower pumps, however, the period through which a pump will safely operate under such a condition is very much of an unknown factor and represents a definite operating hazard which should definitely be avoided. This condition is most likely to occur where the deaerating heater is supplied with bled steam from the main turbine. Sudden load changes, loss of load through electrical disturbance, etc., will cause changes in bled steam pressure which will cause a pressure drop within the deaerating heater. In those cases where two sets of pumps are used in series, flashing on the suction side of the low-pressure pumps will result in flashing in the high-pressure pumps and the same precautions in this regard should be observed for the low-pressure pumps.

Packing

The writer will not attempt to discuss the theories of different types of packings now commercially available for service in high-pressure boiler feed pumps. Rubbing speeds of 70 to 80 ft per sec are common. While the fact is frequently overlooked, stuffing box packing actually forms a bearing and runs with extremely close clearances in order to seal off leakage properly. A "pv" (pressure-velocity) of 700,000 to 1,000,000 is common and such values are several times those allowed in normal bearing practice. In addition to excessively high pv, there is definitely much poorer lubrication, if any, between the stuffing box packing and the shaft sleeve. Aside from the normal pv caused by the hydraulic pressure against the packing, stuffing boxes are subject to wide variations in pressures by adjustments of the gland-bolt nuts against the packing gland. Operators who, through ignorance or carelessness, make frequent adjustments or who practice turning the gland nuts a full turn, can completely wreck a set of packing in a few seconds. After the failure, it is always the packing or the stuffing box that is blamed.

One plant which enjoys real success with high-pressure stuffing boxes has assigned one man on each shift who is the only individual allowed to adjust the gland nuts on stuffing boxes. If the operator feels that the stuffing box needs adjustment, he does not attempt it but calls the man on the shift assigned to this work who decides whether adjustment is necessary. This idea serves the dual purpose of keeping all stuffing box adjustments in the hands of one responsible and experienced person and of eliminating a frequent source of trouble due to the individual operators' varying ideas of proper packing adjustments.

Most commercial packings in use today for high-pressure, high-temperature service are made of materials having a fairly high coefficient of friction. The only way this friction can be reduced is by lubrication and the most logical "lubricant" on boiler-feed service is cold water, either raw filtered water or preferably cold condensate. If the suction pressure is high, it is a common practice to circulate cold water through a lantern gland about in the middle of the box. The inlet cold water is connected to one side of the lantern gland and the outlet to the opposite side. The outlet must be connected to a pressure lower than the inlet to insure cold water getting into the lantern gland. With this arrangement, the hot water being handled by the pump bleeds out through the packing rings between the lantern and the bottom of the stuffing box, mixes with the cold water coming into the lantern, and the mixture goes out the other side of the lantern. This arrangement complicates the situation by requiring the inside rings between the lantern and the bottom of the box to hold usually 50 to 80 per cent of the pressure and none of the cold water gets to these rings to cool or lubricate them. The outside rings between the lantern and the outside of the box may have to hold only 20 to 50 per cent of the pressure and have only cold water against them. This would obviously indicate that the lantern gland should be located at the third ring of packing, in from the outside, so that two-thirds of the packing is used to hold the higher pressure with hotter liquid. The alternative course is to boost the cold water pressure to almost the same amount as the suction pressure and locate the lantern one or two rings away from the bottom of the box. This enables most of the packing to be cooled and lubricated by the cold inlet water.

A method which has met with excellent success is to boost the cold condensate pressure above the inside hot-water pressure against the packing so that cold condensate actually flows into the pump. This is the ideal arrangement as far as packing is concerned as it lubricates and cools each ring of packing. It necessitates a separate source of cold condensate and the pressure from this source must be steady and non-fluctuating. A fluctuating pressure causes compression, decompression, expansion and contraction of the packing which causes a wide fluctuation in drag, pv, lubrication and wear, and results in short packing life. Some operators install an excess-pressure regulator in this line to maintain a constant excess pressure over the suction pressure. Other operators, using the circulating box first described, install a simple constant pressure governor in the cold water inlet line to avoid "weaving" of the packing.

Every attempt should be made to hold the packing pressure at a figure that will insure long packing life and a minimum of maintenance. While this may be less efficient than using a much higher pressure, it will give greater reliability of operation. In addition to keeping the pressure as low as possible, it is highly desirable to have shaft sleeves under the packing made from a tough, hard, wear-resistant polished material to insure a minimum of friction. It is also beneficial to cool and lubricate the packing in every practical way. Generally, the permissible pressure that it is practical to hold will be inversely proportional to the temperature of the liquid in contact with the packing. *(Continued on page 48)*

Burning FUEL OIL and NATURAL GAS

By FRANK G. PHILO

Supt. Steam Power Plants
Southern California Edison Co., Ltd.

These excerpts from a paper[1] presented at the Spring Meeting of the American Society of Mechanical Engineers at Los Angeles, March 23 to 25, 1938, review the practice in gas firing and supplementary oil firing at the Long Beach Station. Aside from this specific information, general observations are made on the handling and burning of these fuels.

THE Long Beach Steam Station of the Southern California Edison Company, Ltd., with a present capacity of 415,000 kw, is equipped for burning natural gas and oil. Prior to 1922, oil fuel was used exclusively but at present it constitutes less than 1 per cent of the annual fuel used.

The normal source of gas supply is the Kettleman Hills field, approximately 208 miles northwest of Long Beach, which is connected with the station by a high-pressure gas-transmission line constructed of 26-in. outside diameter steel pipe with electric-welded joints throughout the greater portion of its length. The gas used is "dry," having previously passed through gasoline-recovery plants. When it leaves the field, its pressure is approximately 400 lb gage, but, before entering the Los Angeles territory, the line passes through a control station at Glendale where the pressure is regulated to give a delivery pressure of about 150 lb gage at Long Beach. On entering the station property, a further reduction to 25 lb is made, after which the gas is metered and distributed to the boiler rooms, being further reduced to a boiler-room supply pressure of 9 lb before entering the buildings.

This arrangement has been satisfactory from an operating standpoint but it is felt that a future installation would carry the 25 lb pressure to each boiler where it would be reduced by individual boiler regulators to 8 or 10 lb, before reaching the burner-header regulating valves. This would insure proper header pressure at all boilers, regardless of changes in loading, and would decrease materially the size of the gas lines within the station.

[1] The complete paper is being published in the April 1938 issue of *Mechanical Engineering*.

Before going to the regulators, the high-pressure gas is passed through a cylindrical separator equipped with suitable traps designed to remove any water or oil. Both gravity and heating values are variable, depending somewhat on each other. Occasionally, gas is received with a heating value as high as 1250 Btu per cu ft, and, at times, it may fall to around 1050 Btu, the yearly average being about 1135 Btu per cu ft.

Each of the six sectional-header boilers in this plant is equipped with 20 combination Peabody oil-and-gas burners of the type shown in Fig. 1, which are designed to deliver 30,000 cu ft of gas per hour each with a pressure at the burner of 3 lb. They are supplied from a 14-in. outside diameter gas header located below the firing floor where the gas pressure is controlled by chronometer valves operated from the Bailey automatic control system. Individual burners are supplied through risers from this header with rising-stem gate valves in the risers immediately above the header and plug cocks on the firing floor for rapid control of the gas to each burner. One of these boilers has been tested at a maximum output of 450,751 lb of steam per hour representing 413.2 per cent of its rated capacity.

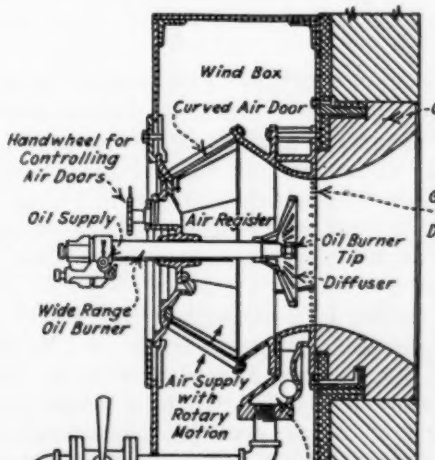


Fig. 1—Combination gas-and-oil burner as installed at Long Beach Steam Station

When burning gas, all burners are fired for all loads, it being possible to reduce the gas input safely to a point where the boilers are generating practically no

steam. Some minor changes in the combustion-control system make possible an increase in the fuel and air supply on the six boilers from minimum to maximum in 45 sec, enabling the 100,000-kw turbine units to accept instantly and maintain a load of 85,000 kw from an initial load of only 1000 kw. While a large number of burners insures even distribution of fuel in the furnace, their use complicates construction of the surrounding furnace lining and multiplies the valves that must be operated when changing fuels. Recently burners of much larger capacity have been developed, and perhaps six burners would be ample for a boiler of this size.

Fuel Oil

Although experience has shown that existing high-pressure gas-transmission lines are reliable and free from trouble, oil is usually used as a reserve fuel and sufficient quantity stored on the premises to protect the station from failure of the gas supply or its diversion to other uses in cold weather. The Long Beach Station is connected by pipe line to a near-by refinery and ordinarily carries sufficient oil in storage for approximately 4 days' supply at normal full load of the 345,000-kw capacity in modern equipment installed in Plants 2 and 3. Because of fire hazard, the storage tanks are usually located as far as possible from other structures and carefully protected.

Fuel oil is ordinarily purchased by the barrel of 42 gal with an approximate average weight of 8 lb per gal or 336 lb per bbl, varying somewhat with the specific gravity. Measurement is referred to a temperature of 60 F. A limit of 2 per cent is usually placed on water and sediment, and it is often specified that the gravity shall not be less than 14 deg A.P.I. at 60 F. Heating value is reported in Btu per pound, a usual value being 18,600 or 6,250,000 Btu per bbl, making 4.5 bbl of oil equivalent to a ton of ordinary coal.

Practically all crude oil is refined. Formerly, the lighter fractions such as gasoline, naphtha and kerosene were distilled off in one large "cut," and the remainder, known as straight-run fuel oil, used as fuel. In later years, the introduction of cracking processes in the refinery has made it possible to rearrange the molecular structure of oils under high temperatures and pressures so that more gasoline and other desired fractions can be recovered, leaving a wide range of products all the way from fuel oil and "Dubbs" residue down through refinery wastes and acid sludge to petroleum coke, all of which are available for fuel either separately or blended to secure certain characteristics. Nearly all fuel oils today contain varying proportions of these cracked residues, and it is a distinct possibility that in the near future all petroleum fuel for industrial use will be 100 per cent cracked fuel.

Accepted maximum viscosities for pumping and atomization are 4000 and 150 sec Saybolt Universal, respectively. In the most commonly used oils, these correspond to temperatures of 80 to 90 F for pumping and 195 to 200 F for atomizing. The general custom is to carry the final oil temperature at between 225 and 250 F.

TABLE 1 ANALYSES AND COMBUSTION CHARACTERISTICS OF REPRESENTATIVE GAS AND OIL FUELS

Gas Analysis

Constituent	Per cent by volume
Methane, CH_4	74.7
Ethane, C_2H_6	23.7
Oxygen, O_2	0.1
Nitrogen, N_2	1.4
Carbon Dioxide, CO_2	0.1
Total	100.0

Heat value at 30 Hg and 60 F, saturated, 1205 Btu per cu ft as determined by Thomas calorimeter. Specific gravity (calculated), 0.681.

Fuel-Oil Analysis

Constituent	Per cent by weight
Carbon, C	86.50
Hydrogen, H_2	11.00
Nitrogen, N_2	0.29
Sulphur, S	0.94
Oxygen, O_2	1.02
Moisture, H_2O	0.20
Ash	0.05
Total	100.00

Heat value, by test, 18,600 Btu per lb. Gravity, 17.9 deg Bé. Flash point, 198 F.

Combustion Characteristics of Gas and Oil Fuels

Constituent	Per cent by weight		Air requirements, lb			Theoretical Combustion Products: CO_2, lb			Moisture, lb			Nitrogen, lb			SO_2, lb			Total
	Gas	Oil	Theoretical per lb of constituent	Per lb of fuel Gas	Per lb of fuel Oil	Per lb of constituent	Per lb of fuel Gas	Per lb of fuel Oil	Per lb of constituent	Per lb of fuel Gas	Per lb of fuel Oil	Per lb of constituent	Per lb of fuel Gas	Per lb of fuel Oil	Per lb of constituent	Per lb of fuel Gas	Per lb of fuel Oil	
C	74.95	86.50	11.52	8.63	9.96	3.67	2.75	3.17	..	..	..	8.85	6.63	7.66	..	..	.	...
H_2	22.72	11.02	34.29	7.79	3.78	..	..	..	8.93	2.03	0.98	26.36	5.99	2.91	..	..	..	...
N_2	2.01	0.29	0.00	0.00	0.00	..	..	..	..	..	..	1.00	0.02	0.00	..	..	..	...
O_2	0.32	1.20	−4.32	−.01	−.05	..	..	..	..	..	..	−3.32	−.01	−.04	..	..	..	...
S	0.00	0.94	4.31	0.00	0.04	..	..	..	..	..	..	3.31	0.00	0.03	2.00	0.00	0.02	..
Ash	0.00	0.05	0.00	0.00	0.00	..	..	..	..	..	..	0.00	0.00	0.00	..	..	..	.
Total	100.00	....	....	16.41	...	..	..	..	..	..	..	....	...	...	..	..	.	17.41
Total	....	....	....	...	...	..	2.75	..	..	2.03	..	....	12.63	...	..	0.00	..	17.41
Total	....	100.00	....	...	13.73	..	..	..	..	..	..	....	..	...	..	..	..	14.73
Total	....	....	..	..	...	..	..	3.17	..	..	0.98	....	...	10.56	.	..	0.02	14.73

It is desirable to provide ample heating capacity for oil in storage, the accepted method being by steam. In cold climates, it may be desirable to insulate the lines between the storage tanks and the plant, and in nearly all cases fuel-oil transfer pumps will be required to speed up delivery even though gravity feed be used from the storage to the burning tanks. While centrifugal pumps are sometimes used for this service, displacement pumps are generally preferred.

Service Tanks

Comparatively small service or burning tanks are usually installed adjacent to the station. These receive their supply from the storage tanks and should be installed in duplicate, each tank holding not less than enough oil for 8 hr full-load operation of the station, oil being withdrawn from one while the other is being filled. These tanks are carefully calibrated and the hourly station use determined therefrom by gaging. Oil in the burning tanks is often heated by returning a portion of the oil as heated for atomization, although heating coils are sometimes used.

Fire protection for both the burning tanks and the

fuel-oil pump rooms is provided by smothering lines supplied with steam at a reduced pressure of 125 lb from the main boiler room. This system also supplies steam to the heaters under thermostatic control for heating the oil to approximately 250 F for the proper atomization. Drips from the heaters are returned through coil drainers to heat exchangers located on the cold oil lines before the heaters and thence to detector tanks, where the drips can be inspected before return to the boiler feed system. Should the drips be contaminated by oil leakage from the heaters, a weir is provided in the detector tank where, by raising the water level slightly, surface accumulations of oil can be skimmed off to the dirty-oil sump.

In the modern central station it is highly important to reduce makeup water requirements. Average makeup requirements in this station for a 75 per cent load factor are about 0.5 per cent of the total boiler feed requirements. Steam required for heating the necessary fuel oil from 70 to 250 F is also approximately 0.5 per cent, and, therefore, makeup requirements would be doubled if the drips from the fuel-oil heater should be discarded.

Burner Supply Lines and Design

Fuel-oil suction lines to the high-pressure pumps should be of ample size. In this station, they are 14 in. and are installed in duplicate. Duplicate simplex

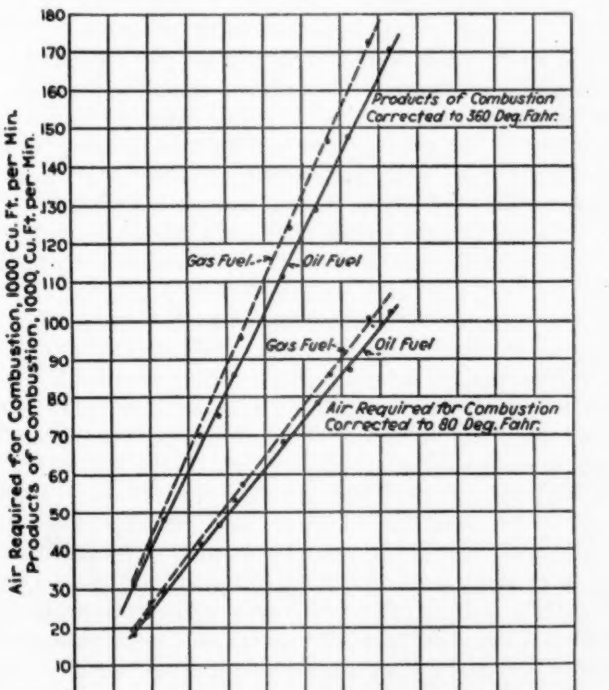


Fig. 2—Air required and products of combustion, oil and gas fuels

strainers are located ahead of each high-pressure fuel-oil pump, which is of the double-screw positive-displacement type and discharges to headers from which the heaters and heat exchangers are manifolded so any piece of equipment can be disconnected without interruption of service. Relief valves on this header are set at 275 lb, and a second, valve-controlled, relief valve set at 50 lb is provided for circulating oil through the system when gas fuel is used.

Early oil burners were of the steam-atomizing type, using steam at about 150 lb pressure and oil at a pressure of approximately 40 lb and a temperature of 140 to 150 F. Although these burners were satisfactory for early boiler installations, their capacity was limited,

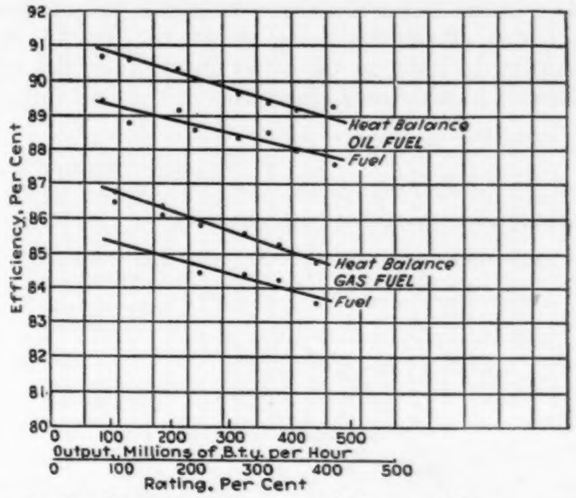


Fig. 3—Combined efficiencies of furnace, boiler, superheater and air heater, oil and gas fuel

and they required approximately 3 per cent of the steam output of the boiler to atomize the oil. They were later almost entirely superseded by the larger capacity and more efficient mechanical-atomizing burner. Recently, a high capacity steam-atomizing burner has been developed which is interchangeable in the same air register with mechanical burners and is reported to use from 0.6 to 1.3 per cent of the steam generated for atomization and to have a wide capacity range. The more commonly used mechanical-atomizing oil burners can be combined in the same air register with natural-gas and pulverized-coal burners, and the fuels can be used interchangeably.

When changing from oil to gas fuel, the oil burners should be either immediately removed and cleaned or thoroughly scavenged with steam or air to prevent accumulation of carbonized deposits in the small holes and slots in the burner tips. When burning other fuels, the oil-burner assembly including the air-diffuser plate is pulled back in the air register to protect it from the heat of the furnace.

Early gas burners were constructed on the principle of the Bunsen burner, a considerable number being located in the furnace floors or walls and the gas being burned at a pressure of 1 lb or less. The modern gas burner usually consists of a circular ring or box placed on the inner side of the air register in which small holes are drilled at a suitable angle to permit the gas to flow into and become thoroughly mixed with the incoming turbulent air stream.

Analyses of representative samples and combustion data for fuel oil and natural gas are given in Table 1. The ratio of carbon to hydrogen is considerably higher in oil than in gas fuel, while the air required and the water vapor formed are considerably less. With the same flue-gas temperature this will show about 4 per

cent greater stack losses for gas than oil. While this should be considered when comparing the purchase price of the two fuels, actual losses in the use of gas fuel above fuel oil are usually less than the theoretical, because gas fuel deposits little or no soot on the boiler heating surfaces, soot blowing is not required and the average heat transfer is better.

Air requirements and products of combustion of oil and gas fuels as secured from test data on the boilers described in this paper are shown in Fig. 2, the comparative tested efficiencies being shown in Fig. 3. It is interesting to note, in the latter illustration, the comparison of heat-balance efficiencies of the two fuels, as

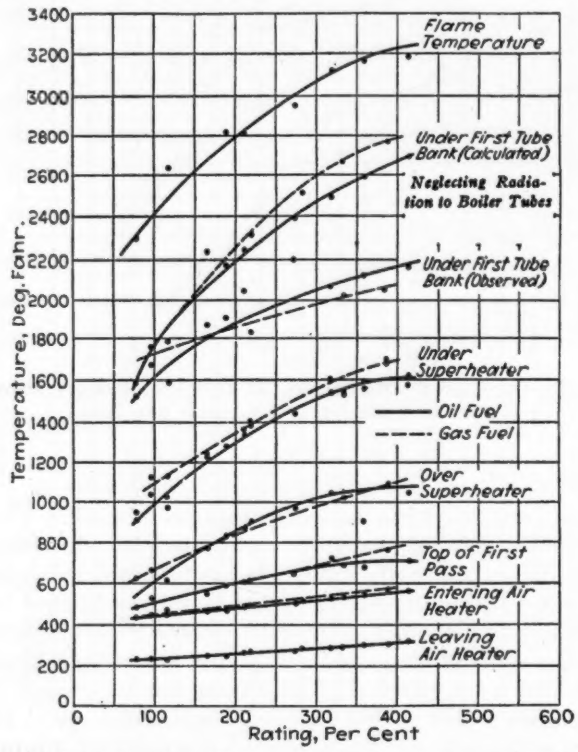


Fig. 4—Distribution of gas temperatures throughout furnace, superheater, boiler and air heater

calculated from their analyses and the exit-gas temperature, with the tested values obtained by weighing the oil and metering the gas. Since the heat-balance method includes all radiation and unaccounted-for losses, it is felt that the average differences of about 1.25 per cent between the two methods represents closely the radiation losses from the boiler tested. With a fuel of uniform composition, the heat-balance method provides a rapid and reasonably accurate method of checking boiler performance at any time.

When burning gas fuel, practically no flame shows in the furnace and temperatures are low, seldom exceeding 2400 F. Refractory maintenance is much reduced, radiant heat is less, and more convection heat is available for the boiler heating surface. Under certain conditions with convection-type superheaters, this will result in higher superheated-steam temperatures with gas than with oil. These relations are shown in Fig. 4.

This boiler room is located on the third floor of the station building, with tiled floors that are regularly waxed and polished, large windows and excellent lighting, and has a general atmosphere of cleanliness and comfort.

Combustible Gas Indicator Used for Detection of Hot Spots in Coal Storage Piles

By GEORGE F. CAMPAU

Technical Engineer, Marysville Power Plant, The Detroit Edison Company

It has been the practice of The Detroit Edison Company for a number of years to make routine surveys of its coal storage piles for detecting hot spots before spontaneous combustion occurs. Until recently at the Marysville Power House, we measured the temperature of the coal by driving a pipe containing a thermocouple into the stock piles and reading the temperature on a portable potentiometer pyrometer. Readings were taken at approximately 20-ft centers and depths of 5, 10, 15, 20 and 25 ft. If temperatures exceeding 145 F were encountered, a survey of the area at this temperature was made on centers of at least 5 ft. Since our storage piles are 30 ft high and contain as much as 130,000 tons of coal, this method was quite laborious and took considerable time. Moreover, it was not always successful in detecting hot spots, as occasionally in reclaiming storage piles a hot spot which we had failed to find was encountered only a few feet away from a thermocouple exploration point.

The labor and time involved and the failure of the thermocouple method always to locate hot spots led us to investigate another method of detection using a portable combustible gas indicator instrument. This equipment consists of a combustible gas indicator calibrated for methane, a $^1/_4$-in. brass sampling pipe and 10 ft of rubber hose. The sampling pipe is closed and pointed at the bottom end to facilitate shoving it into the stock pile, with perforations on the sides near the bottom through which the gas sample is drawn.

Routine surveys of the coal stock piles are made in a manner similar to the thermocouple method except that gas samples are taken at only one depth along the top of the storage piles and tested with the combustible gas indicator. After the indicator has once been adjusted for operation, a survey point can be tested for the presence of combustible gas in a couple of minutes. The instrument is calibrated to read 100 per cent when the mixture of gas and air corresponds to the lower limit of explosibility in air. For methane this is 5 per cent gas and 95 per cent air. It has been our experience that we have found no fire unless a reading of 30 per cent or more of the lower explosive limit was indicated. However, if a reading of 10 per cent or more is indicated, we then explore the area with a thermocouple and try to locate the exact position of the hot spot.

Using the thermocouple in conjunction with the combustible gas indicator may seem paradoxical. This seems necessary, however, since any gas distilled from a hot spot down in the pile seems to diffuse over considerable area. This fact makes it comparatively easy to locate an area containing a hot spot by using a combustible gas indicator, whereas it might be missed if only a thermocouple were employed. The diffusion of the gas is probably due to our not entirely eliminating segregation in building the storage piles and as a result there are seams of coarse coal running up through the piles.

Present Trends of Industrial Power Plant Practice*

By F. H. ROSENCRANTS, Vice President
Combustion Engineering Company, Inc.
New York

The paper deals principally with steam generating units and methods of firing as employed in recent industrial power plant installations. An analysis of about four hundred such installations indicates a trend toward larger units and higher steam pressures, bent-tube boilers, increased use of heat-recovery equipment, and stokers breaking about even with pulverized coal firing. Much of the advancement in industrial power plant practice has been predicated on experience gained in the central station field. Boiler settings, feedwater conditioning and steam purification are also discussed.

AS WOULD naturally be expected, pioneering and development in connection with the use of high-pressure, high-temperature steam has been associated primarily with those undertakings in which the cost of power production represents a major factor in the cost of doing business. It is, therefore, to the public utility field that credit must go for creating the demand for ever-increasing economy in the use of fuel for the production of power. Development of much of the present-day large high-pressure, high-temperature equipment is the direct result of this demand. In the utility field a pressure range of 1200 to 1400 lb has become fairly well established for both condensing and superposed installations, as shown by the number of installations made during the past few years; although there are indications that as new stations are projected many will employ a pressure of 800 to 900 lb and high steam temperatures. In fact, several such plants have been laid down during the past few years.

The situation with regard to steam temperature is less stabilized than that of pressure. As long as the practice of reheating the steam after partial expansion through a high-pressure turbine was in vogue, the temperature was limited to about 800 or 825 F. However, with the obvious desirability of avoiding the complications of the reheating cycle and reluctance to recede from the high pressures already established, the demand for higher steam temperatures to insure a normal dryness of steam through the turbine became insistent. As a result, many of the later central station installations employ temperatures from 900 to 915 F and a few even exceed this. There is a disposition to hesitate at about 900 F, as a maximum steam temperature and there is little, if any, discussion at present of temperatures exceeding 950 F. Further advances will depend upon metallurgical developments and experience.

Large units and varying conditions, making standardization impossible, have characterized recent utility installations.

These brief comments on utility practice with reference to pressure and steam temperature have been made for the purpose of revealing the consequent very firm foundation for the design of steam-generating equipment to satisfy the more moderate demands of the majority of industrial installations. In a few instances the requirements of the industrial plant have been comparable with those of the utilities, but they are the exception rather than the rule. Aside from these few exceptions, industrial requirements fall below 1000 lb pressure and rarely exceed 750 F total steam temperature.

Where steam is required for process operations it is seldom at more than 100 to 150 lb, and frequently less. Where process and heating requirements dictate two or more pressures it is usual to bleed the turbine for steam at the higher pressure or pressures and exhaust at the lower pressure. The justification for generating equipment in excess of these pressures lies, of course, in the value of the power that may be generated by expanding high-pressure superheated steam through the turbine down to the pressure required for process work. The ideal initial pressure is that which will produce the exact power requirements when the quantity of steam demanded by process is expanded through the high-pressure turbine to the process pressure. The corresponding initial steam temperature should be such that after passing through the turbine the steam will issue into the process lines in a saturated condition or with only a few degrees of superheat. For this reason there is not the necessity for high initial steam temperatures such as exists in the straight-condensing cycle.

Of course, there are instances of industrial plants where power production alone is the sole objective and in such cases many factors enter into the decision as to steam pressure and steam temperature. Among these may be mentioned the cost and character of the fuel available, the capacity of the units, the character of the feedwater supply, skill of the operating personnel and its corresponding expense which plant capacity justify.

After steam conditions have thus been tentatively determined it may be found advisable to shade them slightly in order to save the expense of heavier valves and fittings. The A. S. M. E. Boiler Code permits the use of cast-iron feed and blowoff valves for pressures up to 200 lb, above which steel must be employed; whereas bronze valves are allowed up to 240 lb. Steam service valves, including stop valves, are built of steel in 200-,

* Presented at the Midwest Power Conference, Chicago, Ill., April 13 to 15, 1938.

300-, 400-, 600-, 900- and 1500-lb standards. The allowable working pressures, as prescribed by the Code, are modified up or down according to the amount by which the steam temperature is under or exceeds 750 F. Thus a 300-lb valve is good for 390 lb pressure at temperatures not exceeding 450 F and a 600-lb valve is good for 720 lb with temperatures up to 550 F. At 750 F steam temperature the allowable working pressure corresponds with that at which the valve is rated, but with temperatures exceeding 750 F the allowable pressure is less than that stamped on the valve. Safety valves, on the other hand, are rated in accordance with manufacturers' standards and are not modified by temperature.

Trends as Reflected by Recent Installations

With the foregoing points in mind an analysis of contracts for steam-generating equipment, involving approximately four hundred industrial and institutional installations negotiated during 1936 and 1938, are of interest as reflecting present trends. These involved equipment built by a number of manufacturers.

Fig. 1 shows the size of boilers in square feet of heating surface plotted against the number of installations (not individual units). While the points are scattered and do not fall on a curve, it is apparent that there is a marked trend toward units of medium and fairly large capacity. This, of course, does not mean that small units are not being installed where load conditions so dictate, as will be apparent from the considerable number shown on the chart, but rather that the trend for a given plant is toward fewer units of larger capacity, in contrast with the practice some years ago of installing a multiplicity of small units. The improved reliability of steam generating units, their increased efficiency, the lower initial cost of large units for a given steam capacity,

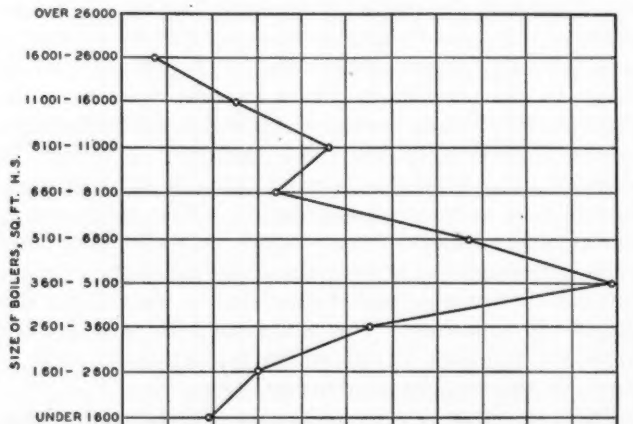


Fig. 1—Installations involving various size units

as compared with two or more smaller units and consequent reduction in operating expense have been responsible for this trend.

Fig. 2 represents the same installations plotted against steam pressure. This shows that approximately half are for pressures in excess of 260 lb per sq in. and over 36 per cent are above 411 lb. Neglecting the few very large units for extremely high pressures, it will be found that the majority of medium capacity units, that is, those from 40,000 to 150,000 lb of steam per hour, fall in the range of 400 to 750 lb, while most of the smaller units are designed to operate at pressures under 260 lb.

Frequently, the purchaser stipulates that the boilers supplied shall contemplate future operation at some considerably higher pressure and temperature than that for

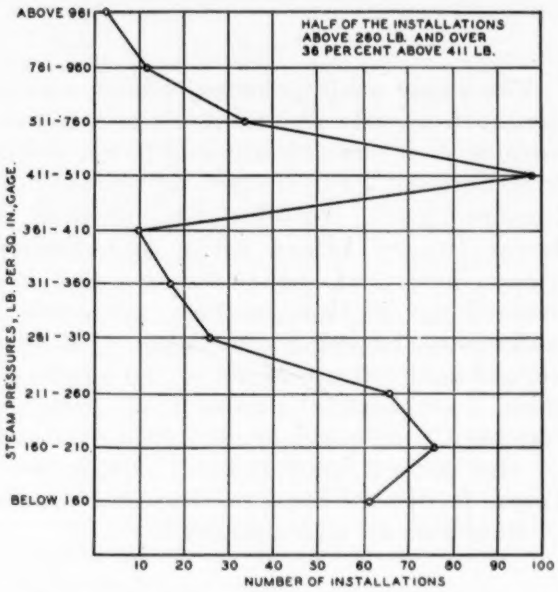


Fig. 2—Installations plotted against pressure

which they are to be placed in operation initially. While this may be a wise precaution in many cases, it nevertheless involves a substantial additional cost over and above the cost of equipment that would satisfy the lower pressure requirements. Although the thickness of all pressure parts, aside from fittings, must be such as to meet Code requirements for the high pressure, the diameter of the steam drums must be greater for the lower pressure because of the greater volume of steam for a given output. As to fittings, they may be selected initially to meet the high-pressure requirements or they may be selected for the lower pressure and changed later. In any event, the boiler will cost more than if it were designed for either low-pressure or high-pressure operation alone. Despite this, the plant may be in a transition stage from low-pressure to high-pressure operation, in which certain low-pressure equipment is still serviceable, and the economics of the problem may fully justify the added cost. Generally speaking, conditions surrounding industrial plants do not warrant superposition to the extent that obtains in many central stations.

Conditions may also dictate the desirability of operating initially with little or no superheat, usually in combination with a pressure increase. Initial superheats of 50 to 100 deg, with provision for future operation at 200 to 300 deg, are quite common. This appears to be a more usual requirement among inquiries received than no superheat initially and provision for the installation of a superheater at some later time. In either case the boiler passes and baffling as originally installed must contemplate the future ultimate requirement and provide the necessary space for the superheater installation. Where only a low degree of superheat is desired initially it is usually wise to lay out the superheater headers so that

they can accommodate the larger superheating surface contemplated for future conditions. The additional cost of such headers is usually small compared with the savings that will result when it becomes necessary to fulfill the ultimate requirements. Where no superheat is initially desired but is contemplated later, it becomes

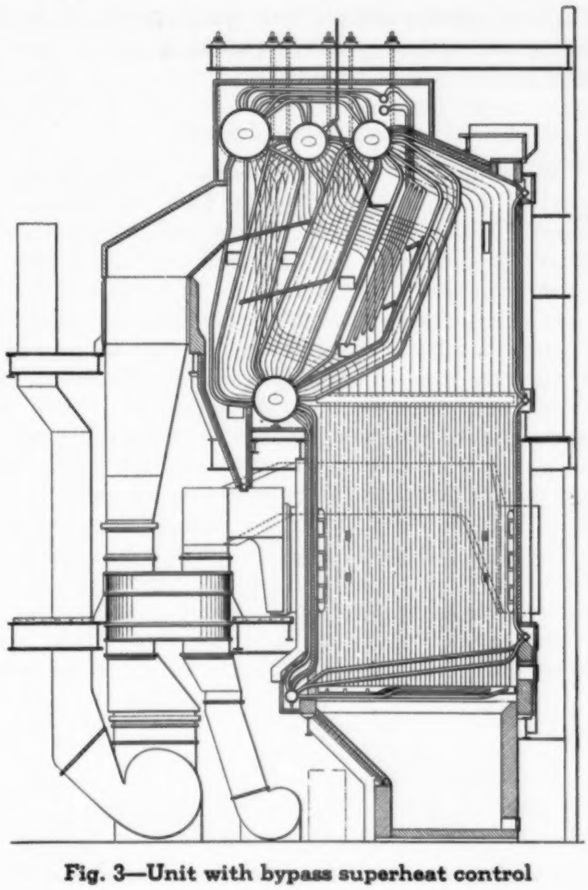


Fig. 3—Unit with bypass superheat control

an open question whether the headers should be installed as part of the original project.

Usually, operating conditions result in a superheat temperature that increases with the steam output of the boiler. In some industrial installations this range of temperature over the operating range of the boiler can be tolerated. Means, however, have been developed for limiting this rise in steam temperature and in many instances the desirability of so limiting and regulating the steam temperature over the upper range of capacity justifies the costs. Numerous schemes have been employed such as spraying water into the steam line, carrying the steam through a surface-type desuperheating device located in the boiler drum, etc.; but bypassing the superheater with a part of the products of combustion has rapidly become accepted as the most desirable and least expensive means. Such an arrangement is provided in most of the large units operating at temperatures around the 900 F mark, and although there are only a few industrial boilers equipped with temperature control devices, the idea seems to be receiving increasing consideration. Fig. 3 shows such an arrangement.

Types of Boilers

The analysis, previously mentioned, showed 82 per cent of the boilers involved to be of the bent-tube type; approximately 40 per cent were fired with pulverized coal; 43 per cent were stoker-fired and 17 per cent burned oil or gas. Economizers were called for in 16 per cent of the cases and air heaters in nearly 40 per cent. The number of air heaters checked very closely with the number of pulverized-coal installations, which was to be expected.

Unlike the utility field, it has been possible to apply a greater degree of standardization in design to the industrial field, although varying steam conditions, fuel characteristics and differences in operating conditions are a bar to complete standardization. For units of 30,000 to 200,000 lb per hr capacity the two-drum bent-tube boiler has found wide acceptance, although in the upper ranges of these capacities, and larger, many of the three- and four-drum bent-tube types have been installed. For the smaller capacities both bent-tube and straight-tube boilers, as well as fire-tube boilers, are being employed. Fig. 4 shows a typical four-drum unit installed in an industrial plant and Fig. 5 is a two-drum unit of a design that has been standardized to a considerable degree.

Water-cooled furnaces have become accepted practice for units of large and medium capacity, particularly where pulverized coal is burned. With stokers complete or partial water cooling is employed according to the conditions and the ignition arches are often water cooled.

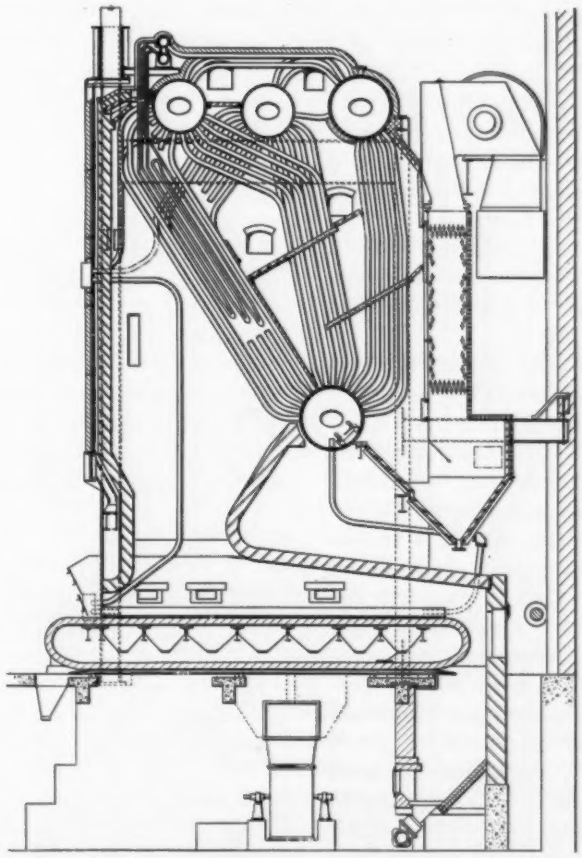


Fig. 4—Typical four-drum stoker-fired unit

For smaller units, particularly those which are stoker-fired, refractory furnaces predominate.

Boiler settings of various types and degrees of construction will be observed among industrial plants. The crudest, perhaps, are those of red brick with a minimum of firebrick lining, inadequate provision for expansion

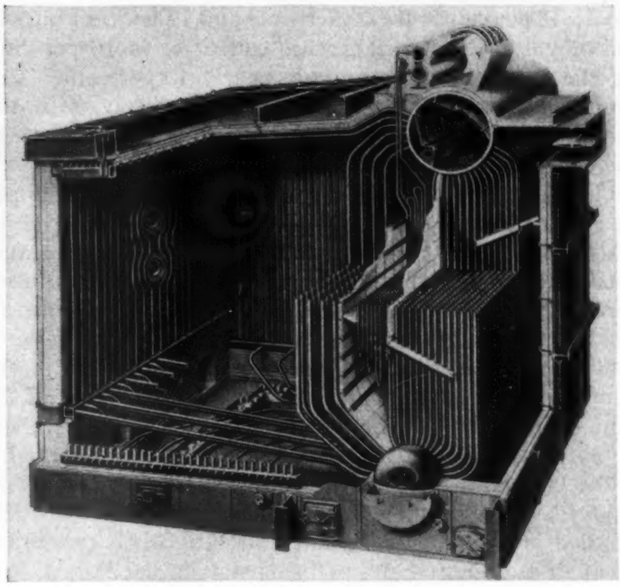


Fig. 5—Two-drum standardized design

joints and poorly buckstayed. Some may be of the same materials but well constructed. Up through the scale of construction will be found settings having various proportions of red brick and first, second and third quality firebrick, or solid firebrick settings with various proportions of first-, second- and third-quality brick. Finally, there are well-constructed firebrick settings covered with insulating material over the entire surface and protected with a tight steel casing. Fig. 6 shows details of such a setting for the two-drum type of boiler illustrated in Fig 5.

While a suitable setting may be had from any of these combinations, if the job is well engineered and properly executed, the author is partial to the best there is in the setting line. A firebrick job that is properly designed and constructed, well insulated and protected with a steel casing should pay dividends over a period of years in a reduction of heat losses. No reasonable amount of brick is equal in insulating value to the proper thickness of a good insulating material of the kind commonly used to insulate boiler settings. On the other hand, deterioration of a steel-cased job, particularly where the furnace is water cooled, is practically nil in comparison with the ordinary run of brick settings. Its permanency and low maintenance merit commendation. A coat of paint applied to the steel-cased setting will restore its appearance and render it comparable in this respect to a new job. The effect of this on the morale of the operators is likely to be worthwhile, whereas the appearance of a brick setting after a few years may be such that neither the operating force, the management nor the manufacturer will be proud of it.

Methods of Firing

The method of firing industrial power boilers is governed to a large extent by local conditions. In certain localities where oil and gas are abundant at attractive prices these fuels receive first consideration. However, inasmuch as a furnace proportioned for oil or gas firing is also well adapted to burning pulverized coal, many such installations are laid out with provisions for changing over to pulverized coal should future conditions make this economically desirable. Such provisions enable the management to take advantage of changing competitive fuel prices and are often most convenient when negotiating contracts for fuel.

Although the majority of large steam-generating units are fired with pulverized coal, stokers and pulverized coal break about even for the medium capacities and the former leads in application to small capacity units of less than 25,000 to 30,000 lb of steam per hour. For caking coal, particularly as mined in the East, the under-

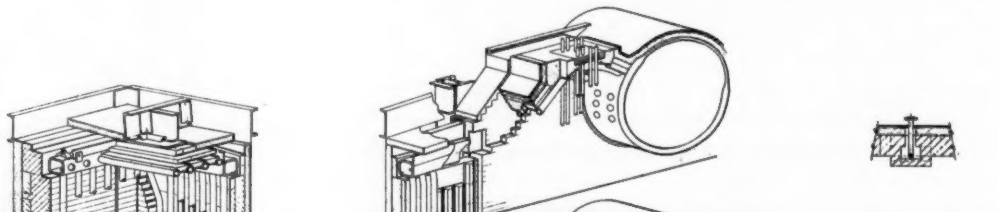


Fig. 6—Steel-cased setting details for unit shown in Fig. 5

feed type of stoker finds its widest application whereas, for free-burning bituminous coals, anthracite and lignite, the traveling-grate stoker is extensively employed. During the past few years the spreader-type stoker has been finding favor in the Middle West. Fig. 7 shows an installation employing this type of stoker.

While it is true that coals covering a wide range in characteristics can be burned more successfully in pulverized form than on any one type of stoker, it is erroneous to assume that all such coals can be burned with equal facility when pulverized. In other words, careful consideration should be given to the selection of fuel-burning equipment in each individual case, regardless of what may appear to be the popular trend.

It occurs to the writer that for the smaller boiler jobs the selection of pulverized fuel equipment does not always take into full consideration the possible hazards involved, and that in many such instances stoker equipment might be a wiser selection. The type of installation referred to is that wherein a single boiler with a single pulverizer may be called upon for considerable periods to operate at light load rating. Under such conditions furnace ignition can be very easily lost due to a variety of causes such as failure of coal supply to the mill, because of the coal hanging up in the bunker, in the spout, in the weighing equipment, in the hopper over the mill or in the feeder. Wet coal or frozen coal will aggravate failure from this cause. Where combustion is under automatic control sudden changes in load at light rating are likely to result in loss of ignition. With the furnace hot, puffs of more or less violence almost invariably ensue upon re-establishing the coal flow and while it is rare that one of these puffs is sufficient to do any damage, there is always the possibility that it may. Under such conditions it seems that stoker equipment, on the whole, would be a better selection.

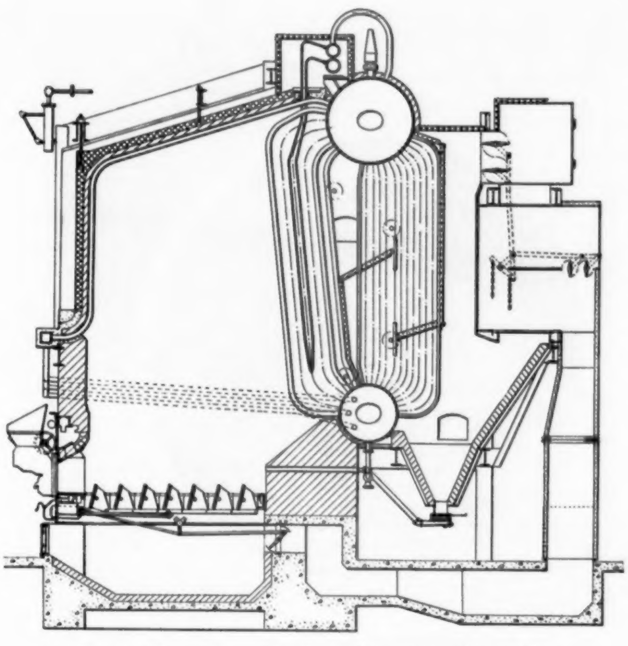


Fig. 7—Two-drum boiler with spreader stoker

As a means of taking care of such interruptions under light load conditions with a single-boiler, single-mill job, an ample size torch fed by either oil or natural gas, depending upon economic availability, is one solution. An alternate suggestion is to insure a steady flow of pulverized coal to the burners by placing a small pulverized coal storage bin with a feeder between the mill and the burner. The bin need not be more than sufficient for a one hour's supply at full capacity. This would last for several hours with the mill shut down entirely and the boiler operating at light load. Aside from the greater reliability which may result, this arrangement permits the adjustment of air to the burner under light load conditions entirely independent of air requirements for proper mill performance. It is the writer's belief that the idea has much to commend it and that the slight additional cost would not be a matter for serious consideration. Otherwise, it is certain that the reliability of the single-mill, single-boiler job as installed for direct firing leaves much to be desired; particularly when called upon to operate under light load conditions.

Inasmuch as the character of the fuel has a direct bearing on furnace proportions and arrangement, in addition to being a determining factor in the selection of fuel-burning equipment, it is important that the steam-generating unit be considered in its entirety when laying out an installation. To this end the consulting engineer or the user's engineer should work in closest harmony with the manufacturer of boiler and fuel-burning equipment and complete information on the characteristics of the fuel to be burned should be available. Where it is anticipated that a considerable range in fuel may be burned during the useful life of the equipment, this should be clearly understood at the start so that the design may take this into consideration.

Feedwater Treatment

Closer attention is now being given to feedwater conditioning for industrial power boilers. This is due not only to the prevalence of higher pressures than formerly, but also to the higher steaming rates now generally employed. Knowledge concerning feedwater and its treatment is being slowly and laboriously accumulated through research and experience. No other phase of practical boiler operation has received so much attention from highly qualified experts. While much pioneer work has gone on in the utility field, industrial plants, because of the high percentage of makeup, often present a more intensified problem. In such cases it is not always economically practical to provide evaporators. Chemical treatment, together with precipitation, may remove certain of the soluble solids and filtering much of the suspended solids. The treated water, however, will still carry a high solid content but of a character more suited to boiler operation than the original water. To maintain the solid content in the boiler water below certain limits requires routine blowdown or, what is better, a continuous blowdown system provided with heat transfer equipment for recovering much of the heat in the blowdown.

Water conditioning is essentially a chemical problem that warrants the guidance of competent specialists in that field. Its proper maintenance, however, is an operating problem in which responsibility rests with the plant management.

So-called "caustic embrittlement," the bugaboo of many years' standing, has been the subject of much

research. Much has been learned as to its cause, but its positive prevention under various conditions of practical boiler operation, as distinguished from laboratory experiments, is yet to be determined. While the subject is one which comes in for much discussion and some argument among experts, they all appear to agree on at least one point, namely, that the effects of caustic embrittlement are decidedly aggravated where the material has been subjected to high stress, that is, a stress above that normally occurring due to the internal pressure only; hence the desirability of maximum freedom for expansion in the matter of restrictions imposed by settings, structural supports, etc. Maintenance of the recommended sulphate-alkalinity ratio has done much to eliminate troubles from embrittlement which were at one time very prevalent; yet despite rigid adherence to this ratio, cases of caustic embrittlement occasionally loom up in specific instances. With the adoption of welded drums, however, the manifestation of damage due to embrittlement is restricted pretty largely to the tube seats, whereas in the days of riveted joints these were the first features to show distress.

Steam Washers

Steam washers are a development that has gone forward at an active pace during the past four or five years. The need for such equipment was emphasized by the difficulties experienced from troublesome deposits on the blades of high-pressure turbines, which often accumulated to such an extent in a few weeks as to cause serious loss in turbine capacity and efficiency. These deposits, of course, come from only one source, namely, solids carried by the steam from the boiler. They may be either water soluble or insoluble. The former are usually found in the high-pressure stages of the turbine and the latter in the low-pressure stages. As a rule, little trouble is experienced at pressures below 300 lb. The usual procedure for the removal of such deposits is to wash them out by periodically passing wet steam through the turbine while the rotor is revolving slowly. This procedure will remove the soluble deposits but to remove insoluble deposits it sometimes becomes necessary to take the machine out of service, lift the cover and mechanically remove the deposits.

The obvious answer is to eliminate the cause of the trouble by eliminating the solids in the steam by means of a steam washer. It should be pointed out here that the steam washer is not a device for reducing the amount of moisture carried from the steam drum to the superheater. The basis of its operation is the substitution of incoming feedwater, of comparatively low solid content, for the relatively high concentrated boiler water carried by the steam. To illustrate, suppose that the water in the boiler has a solid content of 2000 ppm and that the incoming feedwater has a solid content of 200 ppm. In the absence of washing equipment a 1 per cent moisture content in the steam would mean 1 per cent of 2000 or 20 ppm solid carryover in the steam. If now we install washing equipment and substitute incoming feedwater for boiler water as moisture in the steam, the solid content will be 1 per cent of 200 or 2 ppm. Fig. 8 shows one widely used type of steam washer installed in the steam drum in which the steam is washed by bubbling up through the clean feedwater.

Steam washers are, of course, an added auxiliary and as such represent an added expense in the case of boilers. However desirable they may be in conjunction with high-pressure boilers operated in combination with turbine equipment, wherein the water carries solids of a character which will result in troublesome deposits on the turbine blade, there is no justification for their universal adoption. Going to the other end of the scale there certainly is no justification for steam washers in a boiler which is going to operate at any pressure up to 100 or 200 lb per sq in. and which has no superheater and the steam from which is to be used entirely, say, for heating or for process work. For comparatively low pressure turbine operation, that is, 300 lb, the likelihood of any difficulty arising from deposits in the turbine is very much less than is the case in the 500 to 700 lb or above. Furthermore, there are many instances in which the character of the solids is such that they pass

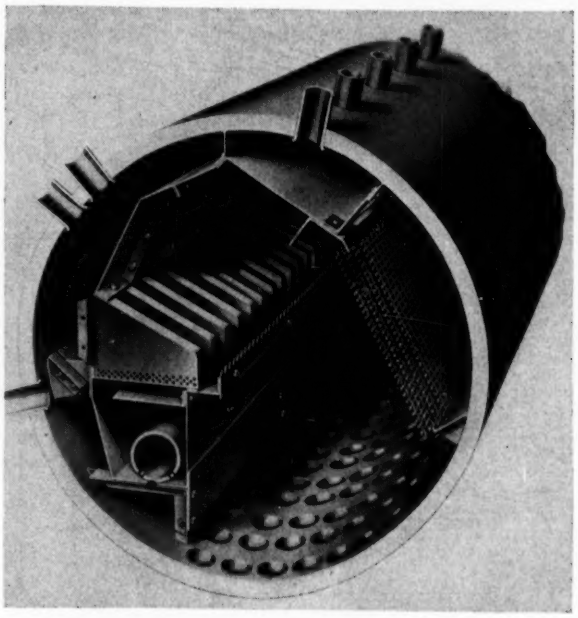


Fig. 8—Bubble-type steam washer

through the superheater and turbine without giving trouble. While all of the elements which will and will not give trouble are far from being completely cataloged, it has been very definitely established that silica in many of its chemically combined forms and sodium hydroxide are among the worst offenders. The foregoing is not intended to be used as a detailed guide concerning the adoption, or otherwise, of washing equipment but is intended only to give an idea of the function of washers and the manner in which they accomplish their objective and their field of usefulness.

Conclusion

In conclusion, attention is directed to the points mentioned earlier in this discourse that the requirements particularly as apply to pressure and temperature, for the normal industrial plant, rest upon a very firm foundation of experience developed in meeting the higher requirements dictated by the utility industry. The industrial field, however, has many problems peculiarly its own associated with water treatment and with widely varying loads and with long periods of light load operation which call for a high degree of judgment on the part of the designing and operating engineers.

Estimation of RADIANT HEAT EXCHANGE in Boiler Furnaces

By GEO. A. ORROK and N. C. ARTSAY

Orrok, Myers & Shoudy
Consulting Engineers,

Engineer,
Foster Wheeler Corp.

This is a rational discussion of the subject in which the results and limitations of various theoretical and mathematical analyses are pointed out. With these as a background, certain modifications are suggested in the light of actual operating conditions and simplified empirical expressions are offered. These are checked with the results of a large number of boiler tests, which are appended.

VALUABLE additional contributions to the knowledge of radiation in boiler furnaces have been made in the last five years. The theoretical and mathematical side of the problem has been perfected by Messrs. Wohlenberg, Hottel, De Baufre and others, and is available through the A. S. M. E. publications. Extended observations are being made on the rates of radiant heat absorption in different parts of furnace envelopes. Litsenberger has offered a simplified method of radiation estimation and Koessler has perfected an experimental apparatus for measuring total radiation in random spots. The work of A. Schack is already well known. The work of developing oil cracking stills indicates the increasing importance of controlled heating of the oil by radiation and extensive tests have been carried out. Wilson, Lobo and Hottel have published[1] a most valuable set of data on radiant heat absorption in oil stills and a comprehensive empirical formulation.

So the observational material accumulates and the mathematical methods are perfected, but still very little use is being made of the latter. There are many instances where large discrepancies between the expected and the actual gas temperatures have been encountered.

It is worth while to review the situation. There appears to be evident reluctance to apply the perfected mathematical methods of radiant heat estimation, but we cannot suppose that this is due to the lack of engineers with sufficient mathematical ability, nor that there is a lack of time for such a study, however laborious it may be. The cause is much more serious, due to the difficulties in applying the known laws of radiation, and misgivings about the assumptions which are usually made or implied.

The radiant heat exchange in boiler furnaces can be estimated in two general ways: First, we may determine radiant heat absorption by cold surfaces and supplement it by calculating the residual sensible heat content in gas and the final temperature of gas at which the latter leaves the furnaces. The second way would be to determine the final gas temperature and sensible heat in it at this thermal level and, by deducting this residual heat in gas from the initial sensible heat content, to obtain the heat absorption by cold surfaces.

By far the most important factor is the furnace temperature, and by this time we are all aware that the measuring or estimating of this temperature is fraught with serious errors. Calculations of this temperature are of doubtful value because we do not know the composition of the gas in different parts of the furnace volume, nor the local rates of combustion. Observed temperatures of flames show such thermal stratification that averaging of temperature in the furnace for use in the Stephan-Bolzman formula is absolutely out of the question. Next come the factors of flame transparency, emissivity and absorptivity. These have been investigated lately and show such large variations that the selection of them is rather a matter of experienced judgment.

Many authors select the outer flame contour as the radiating surface, while being aware that the radiation from flames is in a large measure a volume radiation.

The temperature of water-cooled furnace walls is usually approximated to that of water or steam, but is sometimes neglected entirely as insignificant in the fourth power when compared with the fourth power of flame temperature. The radiation absorbing surface of tubes covered with iron oxide, slag and dust is by no means in good contact with the solid metal of the tube and its temperature may be only a few hundred degrees Fahrenheit below the furnace temperature so that the net radiant heat exchange is strongly affected by the status of this surface. DeBaufre was the first to point this out. Recently Zeuner has reported an experience at the Boehlen Station (Germany). The temperature of the middle metal in the fins of the furnace fin-tube envelope has been measured and for a certain constant rating the re-

[1] *Industrial and Engineering Chemistry*, May 1933.

sults were as follows: Immediately after lighting the fires, 680 C; after six hours operation, 620 C; after four days operation, 400 C.

This temperature variation at the same rating can be explained only by the changed rate of radiation absorption. Saturation temperature being 235 C, we can see that the variation in radiant heat absorption by the fins was of the order of 2.6 to 1.0, all of it being due to the different states of cleanliness of the outer layers on the surface. These temperatures were taken at the middle of the width and thickness of the fins; therefore much higher temperatures should be properly expected on the exposed surface. It has often been observed that new boilers are short of superheat and the usual explanation has been that tubes in the first rows before the superheater are abnormally clean and absorb too much heat when new. A quantitative analysis of this situation would easily show that the cleanliness of the first rows cannot account for the whole effect and some reduction in radiant heat absorption when the surfaces reach a state of "dirt equilibrium" should also be assumed.

The effect of excess air has been introduced in calculations often with much unwarranted refinement. It is obvious that the amount of excess air which is effective in the radiating portions of the flame is by no means equal to that which is measured in the averaged gas. It is quite likely that the intensely radiant parts of the flame are always short of oxygen. The mixing of gases is not instantaneous and in the calculations of total emission from a portion of a flame the temperature if averaged becomes rather an arbitrary figure.

Besides the main and obvious factors influencing the radiant heat exchange, there are a multitude of secondary or doubtful factors. There is a slowly growing realization among engineers that the usual main assumption, namely: that the thermal level of radiation is that of flame or "furnace" temperature, is wrong. A closer study of the basis on which the general (Planck's) law of radiation and its integral, the Stephan-Bolzman formula, are built, would show that the fourth power temperature law does not apply to furnaces where chemical processes of oxidation take place.

It is very curious to note that in the matter of heat transfer between fluids we are already sharply distinguishing the cases where a change of state takes place with its attendant high rates of heat transfer, while the somewhat analogous case of combustion radiation, or "chemiluminescence," is still treated as radiation of solids. There is a possibility that radiation during the chemical reaction might be a large fraction of the total radiation and rather independent of furnace or flame temperature at atmospheric pressure.

The term "furnace temperature" is a very misleading one. Strictly speaking there are several energy levels or equivalent temperatures, for instance, we may conceive first the high levels at which the "chemiluminescence" takes place, then the intermediate levels at which the produced and stabilized molecules of CO, CO_2 and H_2O exist before the next molecular collision, then the gradually lower energy levels of furnace gas, the levels of suspended solid particles in radiant thermal equilibrium with cold walls, and average radiant energy and convection exchange with the surrounding gas. The speed of the processes of solid and gaseous combustion, i.e., generation of energy, is of the same order as that of distribution, and for small groups of a few molecules the equipartition of energy among the degrees of freedom may not exist and the emission of radiation calculated on this assumption may be quantitatively far off.

It is thus advisable *to proceed from the heat balance standpoint* at the furnace exit.

Several years ago, Broido noticed a certain consistency between the fraction of total sensible heat absorbed by radiation and the amount of heat liberated per unit of surface exposed to radiation and drew his curve. At the same time the old Hudson formula was modified to include the most important factors and the following simple expression was obtained:

$$M = \frac{1}{1 + \frac{A}{27} \times \sqrt{C_r}} \qquad (1)$$

where:

M = Fraction of total actually liberated sensible heat absorbed as radiation.
A = Pounds of air per pound coal.
C_r = Pounds of good bituminous coal per square foot of projected surface exposed to radiation.

Some writers have commented on the satisfactory agreement usually obtained when applying this empirical formula, and we are offering a modification of this formula to include a variety of fuels.

$$M = \frac{1}{1 + \frac{A}{25} \sqrt{F_r}} \qquad (2)$$

where: M and A are as before and

F_r = pounds of combustible in the fuel fired per hour per square foot projected radiant surface.

In the case of preheated air the additional heat in the air should be included in the total heat liberated. In the case of gas recirculation as in oil stills, "A" will be total gas per pound of fuel minus one pound.

This expression is purely empirical, giving satisfactory results in the range of gas temperatures 1500 F to 2800 F.

It does not apply for very low furnace loads and also for very high loads when additional phenomena of temporary molecular overexcitation or hidden energy take place. The phenomenon of latent or hidden energy has been investigated by Prof. W. T. David[2] and he found that up to 20% of total generated heat of combustion may be temporarily latent as manifested by shortage of developed pressure during explosions, and in low flame temperatures adjusted for radiation loss. The formula also contains a little inherent divergence in results with fuels high in hydrogen as, for instance, coke oven or natural gas due to relatively high amount of air per pound of fuel, but the error is smaller than the tolerance for tube cleanliness. For these fuels the constant 25 could be increased by 1 for each 5 per cent hydrogen content in the fuel above the 5 per cent usual in coal.

The fraction of total heat absorbed as radiation, usually denoted by M, is a very important item in

[2] *Engineering*, Nov. 12, 1937.

modern boiler design and operation, but when considered broadly and in conjunction with operating conditions which are varying widely for the same boiler, this item is not sufficiently definite to warrant estimation up to, let us say, the third decimal place. So far, we have not yet learned what values of M to select for a boiler design for certain conditions of steaming capacities and fuel properties. It is very difficult to prove the desirability of, for instance, raising M from 0.45 to 0.47 for the normal steaming rate in a boiler, or vice versa. During a year a boiler designed for $M = 0.45$ at normal steaming might have been operated for a large part of the time at two-thirds load; or with such fuels or air ratios that M fluctuated between 0.40 and 0.50. In short, only a reasonable degree of accuracy is actually required of this figure and experience has shown that the simpler the formulas the more use will be made of it. So for all practical purposes an empirical formula for expressing the fraction of the total sensible heat which is absorbed as radiation should be sufficient.

Discussions of radiant heat papers have shown a lack of agreement of the methods of evaluating the effective radiation absorbing surface. The consensus of opinions indicates the desirability of developing an unambiguous uniform method which could make the results obtained by various investigators mutually comparable. This is a difficult problem but at a certain sacrifice of detail a simple plan is workable on the following basis:

(*a*) With the multiple reflection and reverberation of radiant heat in a boiler furnace the effectiveness of the cold surfaces is substantially independent of the solid angle which these surfaces subtend in respect to the center of combustion. In other words, the radiation should be considered largely as incident at right angle to the major plane surfaces.

(*b*) The reflection from cold surfaces is a considerable fraction of total reverberation, hence the banks of tubes should be considered as effective to certain depths.

(*c*) On the strength of (*a*) the basic surface should be taken as a plane or a projection of a curved surface.

Both (*a*) and (*c*) are compromises for the sake of simplicity.

(*d*) Additional surfaces to the main projected planes introduced for reflected radiation incidence should be weighted in proportion to their exposure to total radiation.

The following simplified formulation is then possible:

(1) Plane surfaces, locomotive fire boxes for instance: Full surface taken into account. Net tube sheet surface is increased by the surface inside the fire tubes to the extent of 2 to 3 tube diameters; denoting S_r as effective projected surface, L as exposed length of tubes, d as outside diameter of tubes, and c as distance between tube centers.

(2) Water-wall tubes half imbedded in the refractory close together

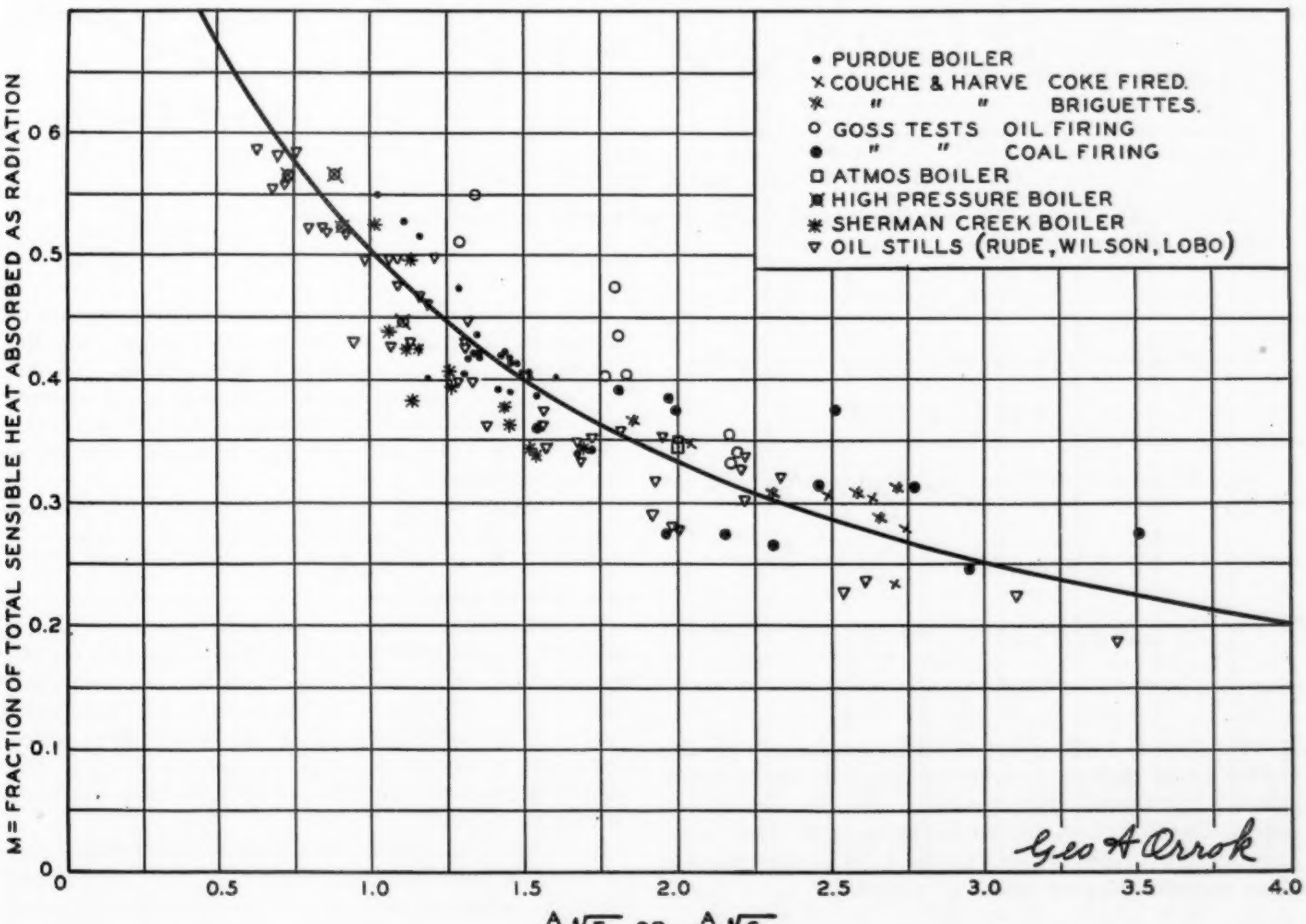


Fig. 1—Values of M from test data plotted against $\frac{A}{25}\sqrt{F_r}$ or $\frac{A}{27}\sqrt{C_r}$

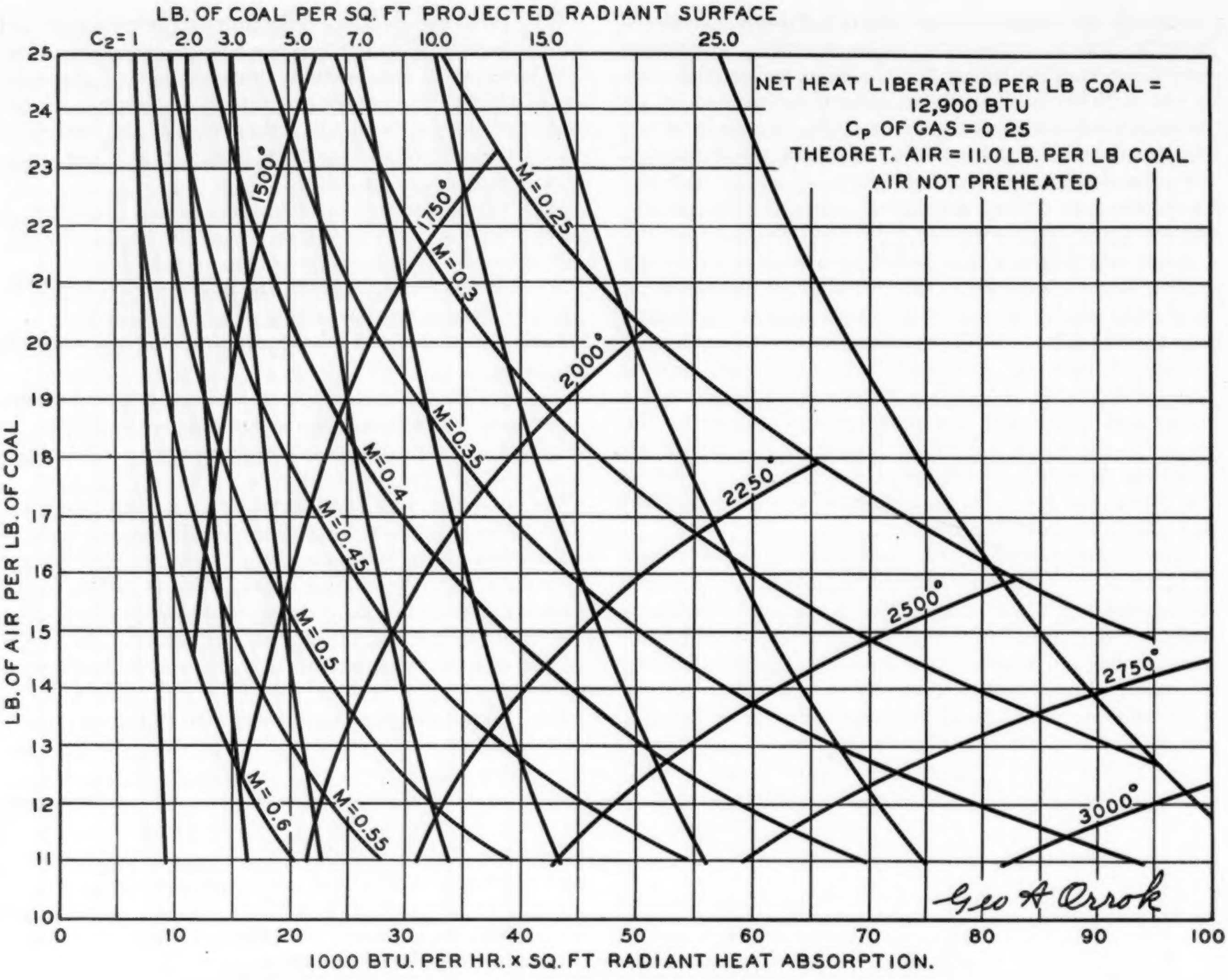


Fig. 2—Air per pound of coal plotted against radiant heat absorption

$$S_r = \Sigma\, Ld$$

(3) Water-wall tubes leaning against the refractory

$$S_r = \Sigma\, Ld\left(1 + \frac{c - d}{2c}\right)$$

(4) Water-wall tubes at a distance from the refactory

$$S_r = \Sigma\, Ld\left(1 + \frac{c - d}{c}\right)$$

(5) Tube banks with more than two rows of tubes

$$S_r = \Sigma\, Ld\left(\frac{c}{d} + \frac{c - d}{c}\right)$$

(6) Water walls covered with refractory blocks. Total plane surface, but the constant in the formulas should be reduced appropriately. In the absence of actual figures of heat absorption by covered water walls a reduction of the constant to one third of its value for bare tubes might give plausible results.

(7) Fin tube and wing-back water walls.

$$S_r = \Sigma\, Lc$$

This method does not claim correct interpretation of the physical meaning of the surface's shape, but the adjustments introduced reflect the general, obvious weight of the partially shielded surfaces and of surfaces receiving the radiation reflected from the cold surfaces.

Some actual test data are here appended with reference to the source.

Fig. 1 shows actual values of M from test data plotted against $\frac{A}{25}\sqrt{F_r}$ or $\frac{A}{27}\sqrt{C_r}$. The curve represents equations (1) and (2).

In general the agreement of actual and calculated values of M is satisfactory, taking into account the roughness of certain tests and the lack of data on combustion losses. The wildness of certain points has been considered in the respective parts of the tables. It is rather surprising to obtain such good agreement while the various degrees of cleanliness of surfaces have not been taken into account.

R. L. Rude has suggested a graphic representation of furnace data derivable from formula (2). Fig. 2 represents the most important factors of boiler furnace operation tied in such a chart. The rate of radiant heat absorption, Btu per hour per square feet (projected) is taken as abscissa, with the total pounds of air per pound of coal

CASE NO. 1

Couche & Havre tests of a locomotive boiler (1874), reported by D. K. Clarke in his book on "The Steam Engine." Radiant surface 76.3 sq ft. Coal briquettes about 13,000 Btu. Sensible heat liberated 92% = 11,950 Btu per lb. Coke = 12,000 Btu. Sensible heat liberated 90% = 10,800 Btu per lb Air per pound of fuel is taken at 20 lb for briquettes, 23 lb for coke.

Test	Fuel	Fuel lb/hr	C_F	Equiv.evap. lb/hr*	Heat lib. 10^6 Btu/hr	Rad. heat absorbed 10^6 Btu/hr	M actual
1	Coke	436	5.7	1820	5.08	1.76	0.347
2	"	655	8.57	2400	7.63	2.33	0.305
3	"	728	9.53	2640	8.48	2.56	0.302
4	"	794	10.4	2650	9.25	2.57	0.278
5	"	772	10.1	2150	9.00	2.09	0.232
6	Briquettes	476	6.23	2145	5.67	2.08	0.367
7	"	743	9.73	2800	8.86	2.72	0.307
8	"	924	12.10	3485	11.00	3.38	0.307
9	"	1025	13.40	3910	12.2	3.80	0.311
10	"	979	12.8	3540	11.65	3.44	0.295

*By radiant surface only

CASE NO. 2

Tests by Prof. W. F. M. Goss of a Jacobs-Shupert and a radial-stay boiler. Projected radiant surface in both boilers, 198.5 sq ft; 34.5 sq ft of tube sheet; total 233 sq ft. The observed firebox evaporation should be corrected for the tube and tube sheet radiant heat by $\frac{198.5 + 34.5}{198.5} = 1.175$.

Heating value of oil = 19,195 Btu/lb. Heat liberated 97% = 18,600 Btu/lb.

OIL FIRING TESTS

Test	Lb oil/hr	Firebox equiv.evap. lb/hr	Heat absorbed, corrected 10^6 Btu/hr	Heat liberated 10^6 Btu	M	F_r	M calc.	
1 J	736	6136	7.0	13.70	0.512	3.16	0.439	air
2 J	1453	10428	11.88	27.10	0.439	6.25	0.358	per
3 J	1399	9227	10.51	25.48	0.404	6.00	0.362	lb
4 J	2156	11683	13.30	39.2	0.340	9.25	0.314	oil =
5 J	2117	11206	12.78	38.5	0.332	9.08	0.316	18.0
6 R	790	6914	7.89	14.37	0.550	3.40	0.429	lb
7 R	1504	9743	11.10	27.40	0.405	6.45	0.354	(25%
8 R	1440	10946	12.48	26.20	0.476	6.18	0.359	excess)
9 R	2106	11942	13.62	38.40	0.355	9.05	0.316	assumed

Remarks—The combustion air was heated in the brick checker work; therefore the actual heat liberated was somewhat larger. The heat in the atomizing steam and heat in the oil also affected the heat liberated. These corrections are not included. In general, the high figures for firebox evaporation in Professor Goss' tests are due to the considerable carryover of water through the steam pipe into the barrel which has not been measured.

COAL FIRING TESTS

Test	Calor. Value	Coal lb/hr	Air, lb per lb of coal	C_F	Combust. losses %	Total heat lib. 10^6 Btu/hr	Equiv. evap. fire-box, cor. lb/hr	Heat abs. 10^6 Btu/hr	M	M calc.
101 J	14380	1368	20.2	5.87	9.7	17.8	7230	7.02	0.394	0.356
102 J	13458	1657	20.0	7.10	8.6	20.35	8170	7.83	0.385	0.337
103 J	14506	2575	20.0	11.05	12.8	32.60	10500	10.18	0.312	0.289
104 J	11967	3090	11.4	13.25	16.1	31.00	11530	11.20	0.361	0.394
105 J	13495	4341	12.3	18.60	15.0	49.70	14100	13.68	0.275	0.338
106 J	13559	4465	14.3	19.15	14.1 assumed	52.00	14170	13.73	0.264	0.302
107 R	13542	1757	24.7	7.55	8.0	21.9	8500	8.25	0.375	0.285
108 R	14604	1452	21.6	6.23	7.8	19.5	7520	7.30	0.375	0.334
109 R	14663	2472	17.9	10.60	10.10	32.4	9100	8.83	0.273	0.317
110 R	13498	2503	21.3	12.45	8.0	36.0	11570	11.20	0.311	0.265
111 R	13076	4345	22.1	18.60	12.0	49.9	14100	13.68	0.274	0.221
112 R	13065	4271	18.7	18.30	8.05 assumed	51.2	12950	12.55	0.245	0.253

CASE NO. 3

Test of an Atmos boiler by Prof. E. Josse (V.D.I. Feb. 14, 1925). Total surface of rotating drums 140 sq ft. Heat absorbed = 5.32×10^6 Btu/hr. Coal fired 1420 lb/hr. Total heat liberated minus losses = 15.5×10^6 Btu/hr. Hence M = 0.344, C_F = 10.1, M calculated = 0.333, A = 17.0.

CASE NO. 4

Sherman Creek boiler No. 10. The projected surface was, side walls, 326 sq ft; superheater, 127 sq ft; burner boxes, 100 sq ft; lower rows of boiler tubes, 135 sq ft. Of the total of 688 sq ft, 553 sq ft were on separate circuits (above 80%) in which the absorption was observed. No data on combustion losses were taken. The projected surface of the tube bank corrected for radiation penetration =

$135\left(1 + \frac{4}{7} \times \frac{7 - 4}{7}\right) = 135 \times 1.57 = 212$ sq ft. Total adjusted surface = 765 sq ft.

Assuming that absorption by the lower boiler tubes per square foot was the same as by the side walls the correction factor would be $\frac{765}{533} = 1.435$. The coal used averaged 14,200 Btu/lb; combustion losses assumed 10%, leaving available heat per pound = 12,800 Btu.

Date	Coal fired lb/hr	A	C_F	Heat liber. 10^6 Btu/hr	Radiation absorbed 10^6 Btu/hr corrected	M	M calc.
Jan. 15	3259	13.2	4.26	41.75	21.9	0.525	0.498
Jan. 14	3260	14.7	4.26	41.75	20.7	0.496	0.472
Feb. 4	3640	13.0	4.75	46.6	19.9	0.427	0.488
Jan. 19	3783	13.5	4.95	48.5	20.6	0.425	0.474
Jan. 18	3800	13.7	4.98	48.7	18.6	0.382	0.470
Feb. 5	4077	13.0	5.32	52.2	22.2	0.425	0.474
Jan. 16	4535	14.0	5.93	58.0	22.9	0.395	0.442
Feb. 6	4960	13.2	6.48	63.6	25.9	0.407	0.445
Jan. 16	5500	14.7	7.20	70.5	25.6	0.363	0.407
Apr. 8	6578	13.2	8.60	84.3	31.8	0.377	0.412
Apr. 6	7939	12.9	10.36	101.6	35.1	0.339	0.395
Apr. 23	7460	13.2	9.70	95.6	33.3	0.348	0.397
Apr. 15	7801	14.3	10.2	100.0	34.3	0.343	0.372

The data assumes dry steam entering the radiant superheater, so actually, due to the presence of moisture in the steam the absorption was greater.

CASE NO. 5

High-pressure boiler. Private information. S_r = 987 sq ft.

Test	Coal fired lb/hr	Heat absorb. 10^6 Btu/hr	C_r	Heat liber.* 10^6 Btu/hr	M	A	M calc.
1	4450	33.3	4.51	58.7	0.567	11.0	0.536
2	2602	19.06	2.64	33.6	0.567	13.0.	0.582
3	4025	24.5	4.08	46.7	0.524	14.0	0.526
4	5215	29.3	5.28	65.4	0.448	15.0	0.475

*Heat liberated includes heat in the preheated air above 80 F

CASE NO. 6

Tests of a high-pressure series steam generator. Reported by A. A. Potter, H. L. Solberg and C. A. Hawkins; A.S.M.E. Annual Meeting Paper 1931.

Total projected surface = 64.3 sq ft furnace walls
9.3 sq ft lower convection bank
73.6 sq ft tubes (corrected)

Correction for heat absorbed in the furnace wall section to get the total radiant heat absorption = $\frac{73.6}{64.3} = 1.143$

Test	Total oil lb/hr	F_r	Rad. absorb. 1000 Btu/hr corrected	Heat liberated 1000 Btu/hr	M	A	M calc.
2	158.7	2.15	1500	2740	0.548	17.3	0.498
3	171.2	2.32	1560	2955	0.528	18.2	0.475
4	191.3	2.60	1700	3290	0.517	17.9	0.465
6	328	4.45	2250	5600	0.402	19.0	0.385
8	387	5.26	2230	6550	0.341	18.2	0.375
9	358	4.86	2100	6110	0.344	19.5	0.368
10	250	3.40	1660	4290	0.388	21.0	0.393
27	189	2.56	1440	3260	0.442	22.1	0.415
29	233	3.16	1700	4010	0.424	20.1	0.412
11	218	2.96	1525	3760	0.405	18.9	0.435
12	209.5	2.84	1430	3570	0.401	17.5	0.459
13	189.5	2.57	1365	3260	0.419	21.1	0.426
14	192	2.60	1385	3280	0.422	20.8	0.428
15	194	2.64	1395	3340	0.418	20.2	0.433
16	235	3.19	1920	4040	0.475	18.1	0.438
17	229	3.11	1720	3940	0.437	19.0	0.427
19	207	2.81	1440	3580	0.402	19.8	0.429
20	216	2.93	1570	3730	0.421	19.7	0.426
21	210	2.85	1535	3630	0.423	20.3	0.422
22	218	2.96	1580	3750	0.421	20.8	0.412
23	224	3.04	1620	3860	0.420	21.0	0.407
24	248	3.37	1670	4270	0.391	19.8	0.408
25	244	3.31	1715	4200	0.413	20.2	0.405
26	241	3.27	1690	4150	0.407	20.6	0.402
32	209	2.84	1410	3600	0.392	22.5	0.398
33	206	2.80	1420	3550	0.400	22.4	0.400
34	209	2.84	1455	3580	0.406	22.5	0.398
35	193	2.62	1383	3340	0.415	22.5	0.408
36	188	2.55	1356	3250	0.417	22.8	0.408

CASE NO. 7

Extensive tests of oil stills, some of which had recirculation of hot gas give the following results. By courtesy of R. L. Rude.

Fuel	Actual fuel lb/hr per sq ft	Heat liber. Btu/hr per sq ft	Air & gas lb/lb fuel	M observ.	M calc.	
						Heat in recircula-
Gas	0.65	25200	106.5	0.186	0.225	tion gas
"	0.75	18800	48.5	0.335	0.374	is added
"	0.70	16600	39.0	0.428	0.434	to heat
"	0.70	16200	34.5	0.469	0.465	liberated
"	0.68	14600	19.0	0.589	0.618	by fuel
"	0.67	14400	21.0	0.584	0.593	
"	0.67	14900	25.5	0.523	0.545	
"	0.66	14900	30.0	0.497	0.507	
"	0.68	16000	39.0	0.400	0.440	
"	0.68	16900	47.0	0.361	0.392	
"	0.73	18900	58.0	0.280	0.336	
Oil	1.01	28500	65.0	0.235	0.277	
Oil	0.95	19700	27.0	0.496	0.489	
Gas	0.8	17300	25.5	0.520	0.525	
"	0.94	20400	21.5	0.520	0.545	
"	0.98	19900	20.0	0.523	0.559	
"	0.88	27100	67.5	0.225	0.283	
"	0.86	18100	19.0	0.558	0.587	
"	1.98	38700	55.0	0.223	0.244	
"	1.88	36500	36.5	0.350	0.333	
"	2.18	42500	33.0	0.353	0.339	
"	2.50	55700	35.0	0.340	0.312	
"	2.86	63600	28.5	0.318	0.342	
"	3.55	69200	31.0	0.322	0.300	
"	4.13	80500	27.0	0.327	0.312	
"	1.03	24300	32.0	0.432	0.435	
"	0.70	14200	36.0	0.500	0.455	
Oil	0.90	17700	28.0	0.430	0.485	
"	1.07	19700	18.0	0.585	0.574	
Gas	0.55	11700	22.5	0.555	0.600	

CASE NO. 8

Reducing furnace 4500 bbl/day capacity. Radiant surface 63 tubes 4 in. OD –, 23 ft exposed. Projected area = $\frac{63 \times 23}{3}$ = 483 sq ft.

Heat in preheated air included in total net heat liberated.

Test	Oil lb/hr	F	A	Sens. heat liberated 1000 Btu/hr	Rad. heat absorbed 1000 Btu/hr	M	M calc.
1	1240	2.56	28.5	24480	8810	0.360	0.355
2	1045	2.16	37.7	20950	6320	0.302	0.310
3	988	2.05	33.5	19735	5760	0.292	0.342
4	1238	2.56	18.4	23390	10780	0.462	0.460
5	1168	2.42	17.3	22110	10560	0.478	0.482
6	1312	2.71	20.1	25230	10120	0.401	0.432
7	1372	2.84	23.2	26295	9910	0.377	0.392
8	1418	2.94	24.4	27550	9680	0.351	0.375
9	921	1.91	36.3	18425	5085	0.276	0.333
10	973	2.02	35.1	19325	5370	0.278	0.335
11	752	1.56	22.4	14510	6215	0.428	0.472
12	728	1.51	19.1	14970	6480	0.433	0.516
13	813	1.69	26.6	16055	5825	0.363	0.420
14	882	1.83	29.0	17360	6005	0.346	0.390
15	1195	2.47	27.3	23520	8290	0.353	0.368
16	980	2.03	18.9	18620	9320	0.500	0.481
17	1057	2.19	22.2	19980	8980	0.450	0.433

as ordinates. The diagram on Fig. 2 is drawn for good bituminous coal of 14,200 Btu per lb (as fired) with 11.0 lb theoretical air requirement. The combustion is assumed with 8 per cent total losses, liberating 12,900 Btu per lb of sensible heat; cold air (80 F) is assumed. Lines for different rates of firing (lb per sq ft) are drawn which indicate the heating effect obtained. The main family of lines is crossed by the lines of equal M and by the lines of equal final gas temperature.

This diagram gives a fairly comprehensive picture of furnace conditions and it is easy to draw for any fuel or combustion air temperature.

The distribution of radiant heat absorption is not uniform throughout the furnace envelope. In most cases we are interested only in the total absorption which allows us to calculate the temperature of gas leaving the furnace, but the increasing use of radiant heat superheaters and reheaters requires a more detailed knowledge of distribution. In this problem very much depends upon the size and shape of the flame. As *a priori* we might say that the lower the value of M (more heat liberated per square foot of radiant surface) the more uniform will be the distribution due to increasing reflection and reradiation. Any attempt to derive a consistent theoretical expression for distribution leads into such a maze of mathematics that the solution would be practically useless. A helpful approximation would be to take the total solid angle B of the projected radiant surfaces (S_r) in respect to flame center and solid angle "b" of the area s_r, which interests us, then to calculate the total radiant heat absorbed H_o in Btu per hr. The absorption H_1 on the selected area s_r then will be a weighted average of that in uniform distribution and in distribution in proportion to solid angles.

$$H_1 = \frac{\frac{Hb}{B}M + \frac{Hs_r}{S_r}}{1 + M} \qquad (3)$$

The importance of solid angles is lesser when M is low, in other words when the rate of firing is higher. This is the highest degree of refinement which appears to be warranted in any formula for radiant heat absorption in furnaces. The radiating heat properties of flames are so indefinite, so changing, and so poorly understood that any further mathematics would be illusory.

Anthracite Conference

The first Anthracite Conference to be held under the auspices of Lehigh University, Bethlehem, Pa., is scheduled for April 29 and 30. Its purpose is to survey recent engineering developments in the mining and utilization of anthracite and to bring engineers, educators, members of the industry and the general public up to date on the rapid technological progress in Pennsylvania's hard coal industry. It is anticipated that the Conference will become an annual affair.

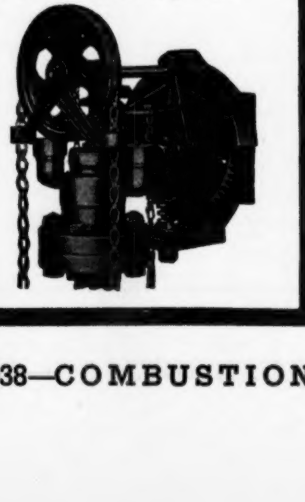
Eighteen papers are scheduled for presentation, the tentative program being as follows:

SESSION I, Friday a.m.
- "Pennsylvania Anthracite Reserves," Dr. Geo. H. Ashley, State Geologist of Pennsylvania
- "Inherent Characteristics of Anthracite," Dr. H. J. Rose, Mellon Institute, Pittsburgh, Pa.
- "Non-Fuel Uses of Anthracite," Prof. H. G. Turner, State College, Pa.
- "A New Theory Concerning the Combustion of Anthracite," Prof. E. S. Sinkinson, Lehigh University
- "Some Practical Considerations in Connection with Combustion," A. J. Johnson

SESSION II, Friday p.m.
- "The Application of Thermostatic Controls to Various Types of Anthracite Equipment," Arnold Michelson
- "Magazine or Self-Feeding Boilers for Coal," Wm. Anderson, Spencer Heater Div., Lycoming Mfg. Co.
- "Domestic Burners (or Stokers) for Anthracite," P. A. Mulcey, Anthracite Industries Laboratory
- "An Improved Method of Hand Firing Pennsylvania Anthracite in Commercial Installations," H. J. Littell
- "Air Conditioning and Refrigeration," Prof. B. H. Jennings
- "Equipment for the Use of Anthracite in Apartment Houses and Semi-Industrial Purposes," Wm. Stein, Combustion Engineering Company, Inc.
- "The Pulverization of Anthracite for Commercial Use," Martin Frisch, Foster-Wheeler Corporation
- "Anthracite for Power Generation," H. W. Warren, Ch. Engr., Glen Alden Coal Company

DINNER MEETING, Friday evening
- Speakers: Dr. A. C. Fieldner, U. S. Bureau of Mines, Washington, D. C. Frank W. Earnest, Pres., Anthracite Industries, Inc., New York City

SESSION III, Saturday a.m.
- "The Distribution of Anthracite," Prof. A Haring
- "Domestic Ash Handling," E. T. Selig, Jr., Mellon Institute
- "The Use of the Degree Day Calculations in the Retail Coal Industry," A. F. Duemler, Household Fuel Corporation
- "The Relation of the Type of Fuel to the Cleanliness of Communities," W. G. Christy
- "Statistical Analysis of the Growth of Pennsylvania Anthracite," Prof. E. C. Bratt



STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Elements and Control of Velox Steam Generator

An article by E. Klingenfuss in *Die Wärme* of December 18 discusses the present status in the development of the Velox steam generator. After reviewing new test data which show that the overall efficiency, including auxiliaries, remains practically constant from $^3/_4$ to $^5/_4$ load, the author describes certain construction details of which the steaming elements and the automatic regulation are of particular interest.

The types of steaming elements used on the latest units are illustrated in Fig. 1. The elements are interchangeable and are removable by loosening four holding bolts at the top and bottom stuffing boxes and water connections. The upper and lower heads are molybdenum-steel castings, the tubes are of the usual boiler tube steel and the assembly is welded into a unit. Use of the different types of elements shown depends upon whether the unit includes a superheater; if it does element *c* is employed, and if it does not either *a* or *b* types are used.

Automatic regulation of the steam generator is illustrated in Fig. 2. It accomplishes a balance between the instantaneous fuel and steam requirements and the steam is replaced with feedwater in such a way that the water content of the generator remains practically constant. In addition there is the regulation of the auxiliary power to the gas turbo-compressor in such a way that at all times, even for sudden load changes, sufficient air is supplied for complete combustion. The fuel control shown at *B* (center) responds to steam pressure, whereas the feedwater control *A* (left) responds to the water level in the steam- and water-separator drum, and the auxiliary load control or mixture regulator *C* (right) responds to the ratio of fuel to air for combustion. All the controls are actuated by oil pressure.

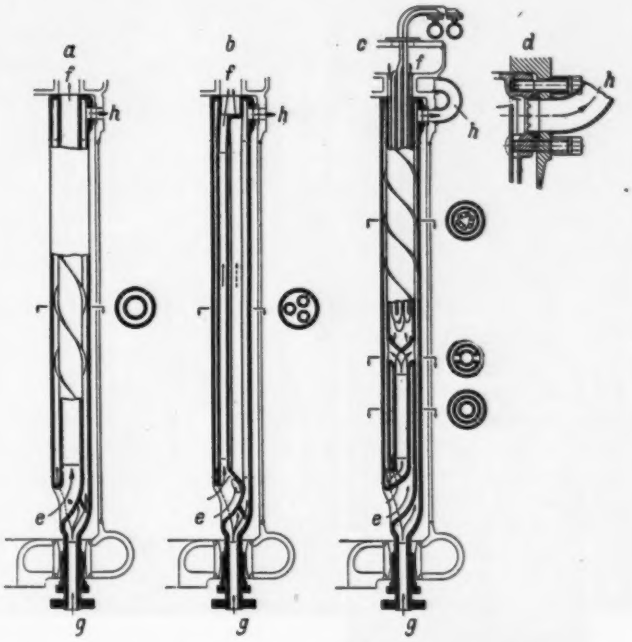


Fig. 1—Different constructions of steaming elements

a, simple element with two concentric tubes; *b*, multitubular element for use with separate superheater; *c*, element with an included superheater; *d*, details of fastening an element to the furnace-generator housing; *e*, gas inlet from the furnace; *f*, gas outlet to the collector casting; *g*, water inlet from the lower water header; *h*, outlet for the water-steam mixture into the upper header

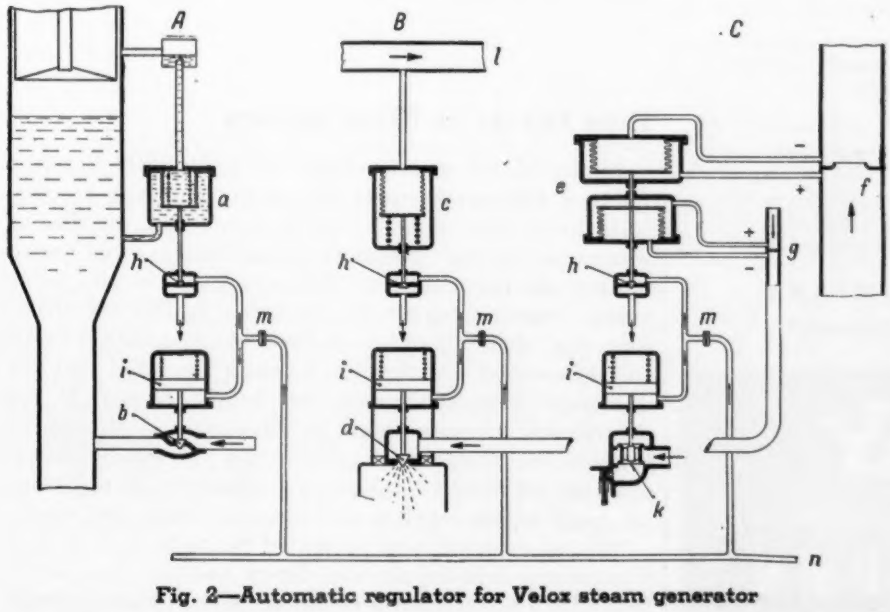


Fig. 2—Automatic regulator for Velox steam generator

For fuel control valve *B*, if the steam pressure in line *l* rises, the bellows *c* drops and the oil valve *h* opens allowing oil to flow from the oil line, thereby reducing the pressure therein and causing the piston *i* to lower and close the fuel valve *d*; *m* is a fixed orifice.

Referring to the feedwater control *A*, if the water level in the drum falls, the bellows *a* sinks, closing the oil valve *h* which throttles the escape of oil from the oil line and thereby increases the pressure therein. This causes piston *i* to rise and open the feed valve *b*. Here also, *m* is a fixed orifice.

For auxiliary power control *C*, the valve in the oil pressure line is connected to two bellows *e*. The

two sides of one bellows are connected to opposite sides of orifice f which measures the air flow. The two sides of the second bellows communicate with opposite sides of orifice g which measures the fuel flow. If air and fuel flow in correct proportions the bellows are in a balanced state. However, if more fuel is fed by control B, a higher pressure will appear on top of the lower bellows, the oil valve will be lower and throttle the escape of oil from the line. This will increase the pressure and cause the piston i to rise and open the throttle valve k which admits more steam to the auxiliary turbine connected to the gas turbo-compressor. Consequently, the compressor speeds up and delivers more air at a higher pressure. The gas turbine now supplies more power and, because of the higher air pressure, the upper bellows moves upward until oil valve h again finds a balanced position. Thus by reduction of oil pressure, the piston i rises and causes the throttle k to reduce or to cut off the auxiliary power.

Small Industrial "Top"

One is accustomed to associate high-pressure topping installations with large or medium size plants, but *Zeitschrift des Vereines deutscher Ingenieure* of December 18 describes a high-pressure extension at the Emil Adolff Works, Reutlingen, in which a Sulzer forced-circulation boiler of 40,000 lb per hr capacity furnishes steam at 1400 lb to two 1200-hp reciprocating engines. These exhaust at a variable back pressure, ranging from 85 to 230 lb, through an oil separator to evaporators and the condensate is returned to the high-pressure boiler. Steam from the evaporators may be used directly for process, in which case the lower back pressure is employed, or it may be used, at higher back pressure, to supply the two 600-hp low-pressure condensing engines, normally served by the old 230-lb boilers. In the latter case it is superheated by passing through a steam reheater supplied with live steam from the high-pressure boiler. Under normal conditions, however, the high-pressure units provide all the power and process requirements of the works.

Pipe Bends as Flow Meters

In the March 4, 1938 issue of *Engineering* (London) Herbert Addison discusses the use of pipe bends to measure the flow of liquids. He observes that the flow of water in curved passages approximates free-vortex motion, the product of the radius and the velocity being nearly constant across the section. It follows, therefore, that there should be a direct relationship between the differential pressure head across the bend and the discharge flowing through the bend. Hence, if the differential pressure head is measured, it should be possible to compute the discharge. A mathematical analysis follows and tables of coefficients of discharge for both square-section and circular bends are given.

The information here presented suggests that there is a promising field for the use of pipe bends to measuring flow with a possible error of not more than plus or minus

5 per cent. It should be particularly useful for determining relative flows. Thus, a pump user who had made a single differential-head reading on a bend in the delivery pipe could tell at any later date whether or not the pump was holding its output. If one single rate of flow could be accurately gaged by other means, the bend could thereafter be used as a meter with a probable error of 2 or 3 per cent, whereas if three points on the calibration curve, spaced well apart, could be established, then the bend should be as reliable as a venturi meter.

The author does not recommend such use of bends if the mean velocity in the pipe is much less than 4 ft per sec principally because the differential head becomes too small for reliable measurement and because of the uncertainty of the value of the coefficient of discharge. Moreover, a sharp bend in which the ratio of curvature radius to pipe diameter radius is less than $2^1/_2$ should be avoided.

The Largest LaMont Steam Generator

The first detailed description of the large LaMont steam generator now being installed in the Deptford West Station of the London Power Company appears in the February issue of *The Fuel Economist.* This unit of 350,000 lb per hr rated capacity (280,000 lb economic rating) is not only the largest of its type but is also the first LaMont forced-circulation boiler to be installed in a public utility power plant. It will be fired by an underfeed stoker and will generate steam at 375 lb pressure and 780 F total steam temperature. Provision has been made, however, for increasing the final steam temperature to 850 F, if desired, by the insertion of additional superheater elements between the first bank of evaporating tubes and the primary superheater.

The unit has a total pressure heating surface of 30,016 sq ft, distributed as follows: Evaporative surface 6639 sq ft, furnace walls 2421 sq ft, superheater 6456 sq ft, economizer 14,500 sq ft; in addition to which there are two air heaters of the regenerative type having a combined surface of 78,600 sq ft. The tubes are $1^1/_2$ in. external diameter throughout, with the wall thickness varying with the location and the duty, and the complex arrangement of the loops has been planned to equalize functional resistance and attain even distribution of the water flowing through them. The total length of tubing exceeds 17 miles.

The combustion chamber is completely water cooled, without any blocks or refractory material except local to the sides of the stoker where abrasion strips are welded on to the tubes.

Three circulating pumps, each of 100 per cent capacity, are provided, two being for standby service. One pump is driven by a steam turbine and two are motor-driven.

In addition to the two induced- and two forced-draft fans there are two secondary air fans, with provision to admit this air at both the front and the back of the combustion chamber.

The unit is provided with a dry drum in addition to the main steam drum.

Business Notes

Edwin W. Nick, president of the Northern Equipment Company, has been named a member of the National Cooperation Committee which is engaged in formulating a program of greater cooperation between industry, the President and Congress.

Pittsburgh Piping and Equipment Company announces the appointment of G. Sinding-Larsen as chief engineer.

The National Aluminate Corporation, Chicago, announces the appointment of W. O. Widener as exclusive representative in the southeastern states, including Virginia, Florida, most of North Carolina, South Carolina and eastern Georgia.

Jones & Laughlin Steel Corporation, Pittsburgh, has appointed three assistant general managers of sales, namely, R. T. Rowles, who will supervise the general activities of the hot-rolled, cold-finished, wire products and warehouse sales departments; W. H. Wiewel, who will supervise by-products, pig iron, sheets and strip and tubular sales departments; and H. J. Watt, who will have charge of sales in New York City and the East, including a new district sales office in Baltimore.

N. B. Ornitz has been appointed president of the Power Piping Division of the Blaw-Knox Company, and **W. N. Quartz** vice president in charge of operations of that division.

W. H. & L. D. Betz, consulting chemical engineers, of Philadelphia, announce the appointment of J. J. McGuire as assistant technical director, and George Bernauer and William Gunmere as members of their Technical Staff. Mr. Bernauer will give attention to the sewage and waste disposal field and Mr. Gunmere will do special service and plant study work.

The Eagle-Picher Sales Company, Cincinnati, has appointed the following as distributors for the Eagle-Picher line of insulating materials: Automatic Coal Burner Company, Beckley, W. Va.; Bemis Refractories Service Company, Springfield, Mass.; The Corbett-Wallace Corporation, Houston, Texas; Joseph A. Coy Company, Tulsa, Okla.; The G. M. Hardware Company, Beaumont, Texas; H. R. Kelly Company, Inc., Brooklyn, N. Y.; Fred H. McGee, Chattanooga, Tenn.; McLean-Peterson Company, Cincinnati, Ohio; United Brick & Tile Company, Oklahoma City, Okla.; United Engineering & Equipment Company, Nashville, Tenn.

Manning, Maxwell & Moore, Inc. is waging a fight against the recession by increasing its general sales force through the addition of nine new field salesmen to strengthen existing territories and to cover new territories with headquarters in Charlotte, N. C., Jacksonville, Fla., New Orleans, La., Minneapolis, Minn. and Columbus, O. Two new district sales managers are also announced as a part of this sales force expansion. They are R. W. Neel, in Atlanta, and Malcolm Black in Tulsa, Okla.

NEW EQUIPMENT

CO_2 Analyzer

The new Pocket CO_2 Indicator, made by the F. W. Dwyer Manufacturing Company, Chicago, constitutes a departure in the instrument field. Complete with carrying case and all accessories, it weighs less than three pounds. It will be a

valuable addition to any boiler room for direct testing of furnace adjustments, and for checking CO_2 recorders for accuracy.

Operation of the indicator is simple, tests being made while it is held in the hand or in the clip provided in the carrying case. The gas sample is pumped in by a dozen strokes of the aspirator bulb while a plunger is held down. The reading is then taken from the mercury column after the absorbing unit has been raised and lowered.

The body of the instrument is made from a new product of the plastics industry which is clear and colorless, and unaffected by the caustic solution used. This is not a molded product, but cut from a solid block with all the bores drilled and reamed to size. There is an entire absence of glass. Despite its small size the analyzer is claimed to be very accurate.

Draft Control

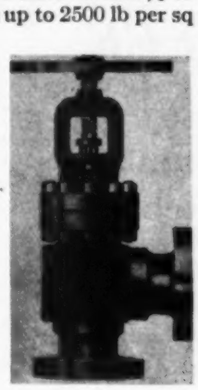
A device known as a Sequence Furnace-Draft Control has been developed by The Hays Corporation, Michigan City, Ind. The primary purpose of this control is to prevent the annoying "blow-back" that occurs at the ignition period in the "on-and-off" type of oil and gas burners and stokers. The control is electrically operated and is precisely timed so as to open the uptake damper wide an instant before the burner or stoker comes on. In this way no pressure is built up in the combustion chamber and the ignition is accomplished without the usual puff.

Immediately following the establishing of proper combustion the damper is closed to the position required to maintain good combustion and from that point on the regular draft control maintains the proper draft.

The advantages of such an instrument can readily be seen. Without such a control the draft is generally higher than required during the "off" period of the stoker-fired boiler, even though it may be near normal during the "on" period. Inefficient combustion results, with consequent fuel losses. The fuel bed is often burned out during the "off" period which follows. The Sequence Draft Control eliminates these losses automatically, by properly regulating the combustion chamber draft to a low and efficient value during the "off" as well as the "on" periods.

Stop Valves

Clees Valve & Engineering Company, New York, N. Y., has brought out a new line of stop valves for high-pressure service. These valves are made from a single block of forged steel in sizes from $2^1/_2$ to 8 in., and for pressures up to 2500 lb per sq in. They can be furnished with flanged ends or with ends prepared for any type of welding. They are also made in both angle and offset types and are produced in conformity with A. S. M. E. specifications.



An unusual feature of the offset valve, which is of the unidirectional type, is that its pressure drop approaches that of a gate valve and is considerably less than the drop in other types. A special internal bypass is provided, which is a desirable feature for the larger valves when used in high-pressure service.

The bodies, flanges and bonnets are all hammer forged from chrome-molybdenum steel, all seats are stellite-faced, bolting is of class "C" A. S. T. M. steel and the trim is of stainless steel.

Nalco Blinker

The Nalco Blinker recently brought out by National Aluminate Corporation, Chicago, is an instrument which acts as a continuous indicator of correct solids content in condensate, steam, feed or boiler water, and for checking the quantity of dissolved solids in any water containing from 1 ppm to 700 gr per gal (12,000 ppm approximately).

The most popular use of the Blinker is to continuously check the solids in the main steam header and thus insure pure steam at all times. This is done by taking a continuous sample of steam through a cooling coil and reducing to a temperature below 100 F. The condensed steam is then passed through an electrode connected by wires to the Blinker which may be in any convenient location.

If the Blinker is connected to a continuous blowdown line, the instrument can be adjusted so that one light glows for too high concentrations and the other for too low. Thus the boiler water solids can be maintained within certain definite limits. The Blinker can indicate solids from only one stationary electrode. If it is desired to have a continuous indication of solids from more than one point, then a Blinker instrument is required for each. Several stationary electrodes may be installed and connected to a ring switch. Solids are then determined periodically by cutting in various electrodes.

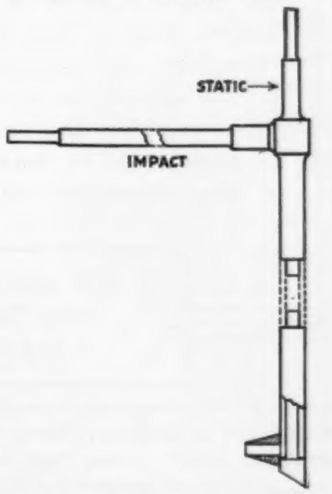
The Blinker instrument measures conductivity which is affected by temperature and difference in type of solids. Temperature is automatically taken care of by a fixed scale behind the rotating dial. No calculating is necessary since solids are read direct. Difference in conductivity, such as caused by an unusually high percentage of sodium chloride or hydrate is compensated for by electrode adjustments and the sensitivity of the lights to changes of solids is regulated.

Liquid Level Gage for Marine Boilers

A modified form of the well-known McNeill liquid level gage, as used extensively in stationary practice, has been developed by the T. W. McNeill Engineering Equipment Company, Chicago, for application to marine boilers which are subjected to the rolling of the ship. By a special construction of the mercury element tilting in any direction does not affect the reading of the instrument. Also, the comparatively large mercury chamber necessary for direct reading is reduced at each extreme so that an undue overbalance of the two connecting columns forces the mercury into these small passages and thus maintains the relative positions of the liquid. By this means, when the true balance is restored, the gage continues to function without the necessity of readjustment.

Pitot Tube

Ellison Draft Gage Company, Chicago, has brought out a new straight-stem pitot tube for use in thick wall ducts where the conventional angle type cannot be used. This is shown in the accompanying sketch.



The outer tube is $^5/_{16}$ in. outside diameter, the flow end of which is machined at an angle of 59 deg, transmitting a true static pressure through the $^1/_8$-in. inner tube. By means of a conical tip, facing the flow, a true impact pressure is transmitted through the outer tube.

Continued from page 26

Maintaining Standby Pumps in Readiness

It is desirable to keep the standby pump always heated approximately to the normal operating temperature to permit its instant use in case of emergency. A small $^1/_2$-in. line connected from the discharge on the pump side of the check valve back to a point of lower pressure than the suction will insure a slow flow through the pump when idle. If the suction pressure is excessively high, the suction valve should be closed and a small bypass line installed around the suction valve. An orifice nipple of suitable size should be installed in the bypass line to break down the pressure and control the flow to the required minimum. If a pressure breakdown device is used ahead of the packing, it is essential to prevent high pressure drop across this during idle periods, as the parts will wear just as rapidly while the pump is idle as though it were in operation. The closed suction valve and bypass are an excellent way to insure minimum pressure drop across these parts. If the suction valve is left open, a valve should be installed in the bleed-off line from the stuffing boxes and this valve kept closed during idle periods. The valve must be opened before the pump is placed in service to insure sufficient fluid flow across the pressure break-down device to prevent over-expansion of the rotating parts causing metallic contact and sudden wear or possibly destruction.

Feedwater Treatment Important

At the time the pump is purchased, the question of the type of feedwater treatment and its relation to materials should be thoroughly investigated. Where the pH value of the water will be definitely maintained above 8.0, pumps should be made of all-ferrous materials, as bronze is subject to caustic corrosion in the higher pH ranges. It is also highly desirable to protect the pre-boiler equipment by the feedwater treatment. Once the plant is in operation, a definite setup should be made to keep the pH of the water the same, within reasonable limits. Proper mechanical deaeration ahead of the feed pumps and after the condenser is usually satisfactory, but many plants are removing the last trace of oxygen by the addition of sodium sulphite. Provided the oxygen content is kept to a trace or less and the pH at the same value, within reasonable limits, no difficulty should be experienced from corrosion provided the proper materials were initially selected for a feedwater of the same pH value with which the pumps are being operated.

Most pump manufacturers engaged in building high-pressure boiler feed pumps have a large amount of experience gained from other installations and it will be to the purchaser's benefit in all cases to consult with the selected manufacturer as to heater locations, pipe sizes, piping methods, proposed operating methods, etc. The suction pressure or effective suction head should be carefully discussed with the pump manufacturer before the final plant layout and flow diagram are definitely settled. This is a most important point affecting pump design and operation and should be accurately and carefully studied in co-operation with the pump manufacturer. This is no place for the station design engineers to arbitrarily decide upon a "factor of safety"

and tell the manufacturer the effective head will only be about half that actually known to exist. The detail piping and layout drawings involving pumping equipment should be given the pump manufacturer for his comments as they pertain to the pumps. The manufacturer, in turn, should make a point of supplying each operator with detailed instructions regarding the pumps. The operators should understand the general features of design and construction so that they may more intelligently understand their job of operation. When the pumps are first placed in service, it is excellent insurance to have a competent service engineer on hand from the pump manufacturer to supervise and instruct the plant operators. Usually two to three weeks will suffice for this period of instruction.

Maintenance Reflects Operating Skill

The maintenance on high-pressure pumps reflects quite accurately the operating skill. Assuming sound basic design and manufacture, high maintenance costs immediately indicate poor or unskilled operation. Here again the pump manufacturer can draw on experiences with similar equipment in other plants and very often will be able to point out effective remedies which will decrease maintenance costs amazingly. If a pump is satisfactorily doing the job it is intended to do, it should not be arbitrarily opened for a scheduled inspection. There are simple ways of determining by outside pressure and flow measurements the approximate internal condition of the pump; and if these indicate the pump is in reasonably good condition, it is always unwise to dismantle it. However, once the pump is dismantled, it is equally good economy to renew any renewable wearing part which shows any sign of wear. It costs an appreciable amount to dismantle and assemble a high-pressure pump and a thorough repair job should be made once it has been decided to spend this money. The first time, at least, that the pump is dismantled, a service engineer from the manufacturer should be employed to supervise the repair work. There are many practical kinks in building pumps which can only be demonstrated by a skilled man and which may very easily be unknown to maintenance men who tackle such a job only once every few years.

And, thus, the *third leg* of our foundation of a successful boiler feed pump installation has been covered. Essentially, it consists of whole-hearted co-operation between the purchaser and pump manufacturer and each should attempt a full understanding of their mutual problems. A good deal of care and thought on the education of new operators will repay itself over and over again and assure the operating supervisors a full night's sleep—in so far as pumps are concerned.

Smoke Prevention Association

May 17 to 20, inclusive, are the dates set for the 32nd Annual Convention of the Smoke Prevention Association at Nashville, Tenn. Headquarters will be at the Andrew Jackson Hotel. In addition to the technical sessions there will be an exhibit of stokers, heating boilers and smoke indicating devices, as well as an educational exhibit featuring various devices used in the abatement of smoke and air pollution, including a display by the U. S. Bureau of Mines.

NEW CATALOGS AND BULLETINS

Any of the publications will be sent on request.

Blow-Off Valve

A bulletin describing for the first time the new Hancock blow-off valve with the unique "blo-deflector" protecting lip has recently been issued by the Hancock Valve Division of Manning, Maxwell and Moore, Inc.

Combustion

The Hays Corporation has recently issued three new publications—one describing its line of draft gages for indicating and recording drafts, pressures, differentials and temperatures; the second, combustion meters; and the third, discusses excess air. The draft-gage bulletin relates the history of the slack leather diaphragm together with its present features and applications. The combustion meter bulletin discusses in a very simple yet comprehensive way the relationship of CO_2 to combustion efficiency and describes in detail the Hays combustion meter which operates on the Orsat principle and uses water as the sole operating power. The bulletin on excess air should be of particular interest to operators of the "on-and-off" type of stoker and oil burner, such as is found in apartment house installations.

Direct-Connected Compressors

Electric-driven direct-connected compressors in sizes from 200 to 3000 hp as built by Ingersoll-Rand Company are featured in a 56-page catalog, No. 3426, just issued by that company. These machines utilize the constant-speed synchronous motor which is efficient at compressor speeds and also valuable in correcting a lagging power factor. The automatic five-step clearance control method of regulation, which cushions the inertia of reciprocating parts, is employed. The catalog is profusely illustrated and incorporates a two-page chart devoted to the interrelation of the compressor design and its operation in service.

Feedwater Control

The Copes "Flowmatic" regulator is described in an 8-page bulletin issued by the Northern Equipment Company. This is a two-element steam-flow type of automatic boiler feedwater regulator designed especially to handle rapid and wide load fluctuations. It is applied with either a direct-operated valve or with an hydraulic-operated valve which is particularly suitable for steam pressures exceeding 600 lb and for large flow through the control valve. Specifications are included.

Filters

Vertical and horizontal pressure filters for clarifying water are described in Bulletin 2760 recently issued by the Cochrane Corporation. These filters are available in either the strainer type or fitted with a single control valve. They are constructed for standard working pressures of 65, 100 and 125 lb per sq in. Their construction and operation are explained in detail and specification tables are included.

Hastelloy Alloys

A new 36-page booklet, "Hastelloy—High-Strength Alloys for Corrosion Resistance," has just been published by Haynes Stellite Company and presents complete information on the four Hastelloy alloys. Points covered in detail in this publication are chemical and physical properties, available forms, methods of fabrication, machining and welding, and typical successful applications. Penetration rates are tabulated for a number of the highly corrosive acids. Information concerning physical and mechanical properties and the available forms in which the materials are furnished will enable the designer to make all necessary calculations.

Pulverized Fuel Sampler

Bulletin No. P-1, issued by The Thorsten Sampler Company, describes a device designed to secure a sample of the coal as carried from the mill to the burner, containing the same proportional amount of coarse and fine material as in the conveying duct. It consists of a sampling spoon which revolves around the centerline of the fuel pipe and is supported on a hollow stand bolted to a base on the pipe. The stand contains the driving mechanism. On the flow face of the sampling spoon is a radial slot with sharp edges divided into two or more openings leading to compartments in the spoon; these terminate in a plenum chamber which communicates with the hollow stand and, through a shut-off valve, with the sample bag. The sampler can be used for any material, other than coal, that can be transported through a circular duct by the pneumatic method.

Pumps

The new East Park Pumping Station of the City of Philadelphia is described and illustrated in a leaflet distributed by the De Laval Steam Turbine Co., builders of the pumping equipment. This new station is located alongside a 48-in. cast-iron main feeding into the main section of Philadelphia and was designed to improve the conditions in the large downtown or central section of the city, to supply the excess demand of South Philadelphia in extremely hot periods, and to act as a standby to supply the West Philadelphia area in case of trouble at the Belmont Pumping Station. The three pumping units are driven by synchronous motors and are each designed to deliver 25 million gallons per day against 149 ft head, or 20 million gallons per day against 170 ft head. Actual test after installation showed an efficiency from wire to water exceeding 85 per cent, and one of the pumps, delivering at the rate of 25.01 mgd from 13.4-ft suction head to 157.9-ft discharge head, showed 90.8 per cent pump efficiency.

Regulators

Publication 34-9-3 is a condensed catalog issued by the Spence Engineering Company, Inc., including, among a long list of products, back-pressure valves, barometric regulators, combustion control regulators, differential pressure regulators, desuperheaters, excess-pressure pump governors, fan-engine regulators, fuel-oil pump variable-pressure governors, strainers, control for heating systems, etc. Tables of dimensions and descriptive information are included, as well as engineering data in the form of charts and curves.

Steam Traps

A 12-page bulletin, T-1733, just issued by Yarnall-Waring Company, describes the Yarway Impulse Steam Trap. Construction details are fully illustrated and described and its applications discussed. A convenient abbreviated table of thermodynamic properties of saturated steam is included and trap specifications covering capacities, weights, dimensions and prices are given.

Water-Vapor Refrigeration

The development, perfection and advantages of the centrifugal type of water-vapor refrigeration as well as typical installations are presented in a new 32-page booklet issued by Ingersoll-Rand. Using water as the only refrigerant, these units represent the latest development in cooling water for air conditioning and a variety of applications in industrial and chemical plants. The units can be driven by electric motor or steam turbine to economically meet any condition. Standard dimensions, as well as illustrations of diversified typical applications, are shown in the bulletin.

Wrought Iron

One hundred and one uses for wrought iron are discussed and fully illustrated in a 32-page booklet issued by A. M. Byers Company. While these uses cover a very wide field there are a number applicable to the power plant.

EQUIPMENT SALES

Boiler, Stoker, Pulverized Fuel

as reported by equipment manufacturers of the Department of Commerce, Bureau of the Census

Boiler Sales

	1938 Water Tube		1937 Water Tube		1938 Fire Tube		1937 Fire Tube	
	No.	Sq Ft	No.	Sq Ft	No.	Sq Ft	No.	Sq Ft
Jan.	52	201,151	52	256,368	35	42,752	65	84,889
Feb.	48	185,257	51	198,957	45	55,173	74	89,133
Jan. to Feb. Inclusive	100	386,408	103	455,325	80	97,925	139	174,022
1937—12 mos.			861	4,058,481			941	1,178,805

Mechanical Stoker* Sales

	1938 Water Tube		1937 Water Tube		1938 Fire Tube		1937 Fire Tube	
	No.	Hp	No.	Hp	No.	Hp	No.	Hp
Jan.	28	9,484	63	25,278	76	10,991	140	21,636
Feb.	34	12,002	45	16,591	75	12,166	120	20,650
Jan. to Feb. Inclusive	62	21,486	107‡	41,719‡	151	23,157	261‡	42,436‡
1937—12 mos.			659	262,834			2,628	361,346

* Capacity over 300 lb of coal per hour
‡ Corrected

Pulverizer Sales

	1938 Water Tube			1937 Water Tube			1938 Fire Tube			1937 Fire Tube		
	No.		Cap. Lb	No.		Cap. Lb	No.		Cap. Lb	No.		Cap. Lb
	N.†	E.†	Coal/Hr	N.†	E.†	Coal/Hr	N.†	E.†	Coal/Hr	N.†	E.†	Coal/Hr
Jan.	5	—	40,500	35	7	554,900	1	—	1,000	—	2	1,700
Feb.	7	1	38,020	2	6	68,300	—	1	800	—	4	3,600
Jan. to Feb. Inclusive	12	1	78,520	37	13	623,200	1	1	1,800	—	6	5,300
1937—12 mos.				214	65	2,924,590				3	10	11,900

† N—New boilers E—Existing boilers.

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